10/085,112

DECLARATION FOR TRANSLATION

I, Jun Ishida, a Patent Attorney, of 1-34-12 Kichijoji-Honcho, Musashino-shi, Tokyo, Japan, do solemnly and sincerely declare that I well understand the Japanese and English languages and that the attached English version is a full, true and faithful translation made by me

this 11th day of June, 2004

of the Japanese priority document of

Japanese Patent Application
No. 2001-58513

Entitled "PULLEY THRUST CONTROL DEVICE FOR BELT-TYPE CONTINUOUSLY VARIABLE TRANSMISSION UNIT"

In testimony thereof, I have herein set my name and seal

this 11th day of June, 2004

Jun Ishida

[Name of Documents] APPLICATION FOR PATENT

[Identification No. of Documents] TC1-0512

[Filing Date] March 2, 2003

[Addressee] Esq. Commissioner of the Patent Office

[IPC] F16H 9/18

[Inventor]
[Address] c/o KABUSHIKI KAISHA TOYOTA CHUO KENKYUSHO 41-1, Aza Yokomichi, Oaza Nagakute, Nagakute-cho, Aichi-gun, Aichi-ken, Japan
[Name] Masatoshi HADA

[Inventor]
[Address] c/o KABUSHIKI KAISHA TOYOTA CHUO KENKYUSHO 41-1, Aza Yokomichi, Oaza Nagakute, Nagakute-cho, Aichi-gun, Aichi-ken, Japan
[Name] Hideyuki SUZUKI

[Inventor]
[Address] c/o KABUSHIKI KAISHA TOYOTA CHUO KENKYUSHO 41-1, Aza Yokomichi, Oaza Nagakute, Nagakute-cho, Aichi-gun, Aichi-ken, Japan
[Name] Hiroyuki NISHIZAWA

[Inventor]
[Address] c/o KABUSHIKI KAISHA TOYOTA CHUO KENKYUSHO 41-1, Aza Yokomichi, Oaza Nagakute, Nagakute-cho, Aichi-gun, Aichi-ken, Japan
[Name] Ichiro TARUTANI

[Inventor]
[Address] c/o KABUSHIKI KAISHA TOYOTA CHUO KENKYUSHO 41-1, Aza Yokomichi, Oaza Nagakute, Nagakute-cho, Aichi-gun, Aichi-ken, Japan
[Name] Yuji NAGASAWA

[Inventor]
[Address] c/o KABUSHIKI KAISHA TOYOTA CHUO KENKYUSHO 41-1, Aza Yokomichi, Oaza Nagakute, Nagakute-cho, Aichi-gun, Aichi-ken, Japan
[Name] Masataka OSAWA

[Inventor]
[Address] c/o TOYOTA JIDOSHA KABUSHIKI KAISHA
1, Toyota-cho, Toyota-shi, Aichi-ken, Japan
[Name] Kunihiro IWATUKI

[Inventor]
[Address] c/o TOYOTA JIDOSHA KABUSHIKI KAISHA
1, Toyota-cho, Toyota-shi, Aichi-ken, Japan
[Name] Yasunori NAKAWAKI

[Applicant]
[Identification No. of Applicant] 000003609
[Name] KABUSHIKI KAISHA TOYOTA CHUO KENKYUSHO

[Applicant]
[Identification No. of Applicant] 000003207
[Name] TOYOTA JIDOSHA KABUSHIKI KAISHA

[Agent]
[Identification No. of Attorney] 100075258
[Patent Attorney]
[Name] Kenji YOSHIDA
[Telephone Number] 0422-21-2340

[Elected Agent]
[Identification No. of Attorney] 100081503
[Patent Attorney]
[Name] Toshihiko KANAYAMA
[Telephone Number] 0422-21-2340

[Elected Agent]
[Identification No. of Attorney] 100096976
[Patent Attorney]
[Name] Jun ISHIDA
[Telephone Number] 0422-21-2340

[Official Fee]
[Registered No. for Payment] 001753
[Amount] 21000

[List of Filing Papers]
[Name of Item] Specification 1
[Name of Item] Drawings 1
[Name of Item] Abstract 1

[Necessity of Proof] YES

[Name of Document] Specification

[Title of the Invention] PULLEY THRUST CONTROL DEVICE FOR BELT-TYPE CONTINUOUSLY VARIABLE TRANSMISSION UNIT

[Claims]

1. A pulley thrust control device for a belt-type continuously variable transmission unit comprising a driving pulley and a following pulley connected via a belt with the driving pulley, and capable of continuously changing a speed changing ratio by changing effective diameters of the driving pulley and the following pulley,

wherein a thrust ratio between the thrust of the driving pulley and the thrust of the following pulley is determined, and

pulley thrust is controlled based on a state of change of the thrust ratio.

2. The pulley thrust control device for a belt-type continuously variable transmission unit according to claim 1, wherein the pulley thrust is controlled such that the thrust ratio approaches a point at which the gradient of change of the thrust ratio changes.

3. The pulley thrust control device for a belt-type continuously variable transmission unit according to claim 1 or 2, wherein the state of change of the thrust ratio is determined while the pulley thrust is varied according to a predetermined cycle.

4. The pulley thrust control device for a belt-type continuously variable transmission unit according to any one of claims 1 to 3, wherein the thrust ratio is determined by measuring a hydraulic pressure which controls thrust of the driving pulley and the following pulley.

5. The pulley thrust control device for a belt-type continuously variable transmission unit according to any one of claims 1 to 3, wherein the thrust ratio is determined based on a command value for a hydraulic pressure which controls thrust of the driving pulley and the following pulley.

6. The pulley thrust control device for a belt-type continuously variable transmission unit according to any one of claims 1 to 5, wherein an average friction coefficient ratio is used in place of the thrust ratio so that the pulley thrust is controlled based on the state of change of the average friction coefficient ratio, the average friction coefficient ratio being obtained by multiplying the thrust ratio by a ratio between belt hanging diameters of the driving pulley and the following pulley.

7. A pulley thrust control device for a belt-type continuously variable transmission unit comprising a driving pulley and a following pulley connected via a belt with the driving pulley, and capable of continuously changing a speed changing ratio by changing effective diameters of the driving pulley and the

following pulley,

wherein a control map for controlling pulley thrust is amended based on a state of change of the thrust ratio.

[Detailed Explanation of the Invention]

[0001]

[Field of the Invention]

The present invention relates to a pulley thrust control device for a belt-type continuously variable transmission unit which comprises a driving pulley (primary pulley) and a following pulley (secondary pulley) connected to each other by a belt and which allows continuous changing of a speed changing ratio by changing the effective diameters of both of the pulleys. In particular, the present invention relates to control of thrust or a belt clamping force of the pulleys.

[0002]

[Description of the Prior Art]

Conventionally, continuously variable transmission units capable of continuous changing of a speed changing ratio have been known for use as a power transmission unit for vehicles. As such a continuously variable transmission unit, a belt-type continuously variable transmission unit in which a driving pulley (primary pulley) and a following pulley (secondary pulley) are connected to each other via a belt and the effective diameters of the driving and following pulleys are changed is widely employed.

[0003]

In such a belt-type continuously variable transmission unit, substantially conic sheaves are opposed to form a pulley and the distance between the sheaves is changed to thereby change the effective diameter of the pulley. In order to change the

effective diameter of the pulley, most commonly, the sheaves are hydraulically driven. Thus, the belt clamping force of a pulley (pulley thrust) is hydraulically controlled. It should be noted that belts in common use comprise a number of blocks fixed by a strip-like hoop.

[0004]

In such a belt-type continuously variable transmission unit, the thrust of one of the pulleys (for example, the driving pulley) is determined to determine the speed changing ratio, while the pulley thrust of the other pulley (for example, the following pulley) is determined such that no slip occurs.

[0005]

Although belt slip could be reliably prevented if the pulley thrust of the following pulley is set sufficiently large, this may cause a problem that power transmission efficiency may be deteriorated. When, on the other hand, the pulley thrust is set too small, belt slip may occur, which further causes a problem that sufficient power transmission cannot be carried out.

[0006]

In other words, as shown in Fig. 22, increasing the ratio of transmission torque to transmission tolerance torque (transmission torque/transmission tolerance torque) leads to an increase of transmission efficiency and also a gradual increase of a belt slip ratio, the transmission torque being the torque actually transmitted and the transmission tolerance torque being a torque transmittable without causing belt slip.

In addition, when this ratio approaches 1.0, such characteristics are presented that the belt slip ratio sharply increases, causing macro-slip and a corresponding drop in transmission efficiency.

[0007]

Conventionally, belt slip is detected to set the pulley thrust such that the belt slip is limited to a predetermined amount. This makes it possible to suppress belt slip and improve transmission efficiency.

[0008]

[Problem to be solved by the Invention]

However, because such conventional pulley thrust control reacts to observed belt slip itself, a certain amount of slip is to be allowed. As a result, there was a problem in that disturbances, such as a large change in pulley transmission torque or the like, often cause large belt slip (macro-slip).

[0009]

The present invention was conceived to solve the above-described problem and aims to provide a pulley thrust control device for a belt-type continuously variable transmission unit, which can appropriately control a pulley thrust.

[0010]

[Means for solving the Problem]

The present invention is characterized in that a pulley thrust control device for a belt-type continuously variable transmission unit comprises a driving pulley and a following pulley connected via a belt with the driving pulley and is capable

of continuously changing a speed changing ratio by changing effective diameters of the driving pulley and the following pulley, wherein a thrust ratio between the thrust of the driving pulley and the thrust of the following pulley is determined, and pulley thrust is controlled based on a state of change of the thrust ratio.

[0011]

The thrust ratio peaks immediately before a large belt slip (macro-slip) occurs and power transmission efficiency is maximized also immediately before macro-slip occurs. Thus, appropriate control of the pulley thrust can be performed by controlling the pulley thrust in accordance with the state of change of the thrust ratio.

[0012]

Further, it is preferable that the pulley thrust is controlled so as to closely approach a point at which the gradient of changing of the thrust ratio changes. The thrust ratio peaks immediately before macro-slip occurs and the maximum point of power transmission efficiency exists also immediately before macro-slip occurs. Thus, such control enables appropriate control of the pulley thrust.

[0013]

Further, it is preferable that the state of change of the thrust ratio is determined while the pulley thrust is varied according to a predetermined cycle. Thus, by periodically changing the pulley thrust, the peak of a thrust ratio can be easily detected.

[0014]

Further, it is preferable that the thrust ratio is determined by measuring a hydraulic pressure which controls thrust of the driving pulley and the following pulley. By measuring a hydraulic pressure, pulley thrust can be easily measured.

[0015]

Further, excess thrust can be easily detected based on the state of change of a phase of the thrust ratio with respect to the following pulley thrust.

[0016]

Further, it is preferable that the thrust ratio is determined based on a command value for a hydraulic pressure which controls thrust of the driving pulley and the following pulley. Thus, determination means such as a hydraulic sensor can be omitted.

[0017]

Further, it is preferable that an average friction coefficient ratio is used in place of the thrust ratio so that the pulley thrust is controlled based on the state of change of the average friction coefficient ratio, the average friction coefficient ratio being obtained by multiplying the thrust ratio by a ratio between belt hanging diameters of the driving pulley and the following pulley.

[0018]

Because the average friction coefficient ratio changes according to the speed ratio, suitable thrust control can be

performed even though the speed changing ratio changes.

[0019]

Further, the present invention is characterized in that a pulley thrust control device for a belt-type continuously variable transmission unit comprises a driving pulley and a following pulley connected via a belt with the driving pulley and is capable of continuously changing a speed changing ratio by changing effective diameters of the driving pulley and the following pulley, wherein a control map for controlling pulley thrust is amended based on a state of change of the thrust ratio. Thus, optimal thrust control can always be performed.

[0020]

[Preferred Embodiments]

In the following, preferred embodiments of the present invention will be described based on the drawings.

[0021]

"First Embodiment"

Fig. 1 is a diagram showing a complete structure of a first embodiment. An input shaft 10 from an engine is connected to a driving pulley 12, which consists of sheaves 12a, 12b. The driving pulley 12 comprises a fixed sheave 12a and a movable sheave 12b, the movable sheave 12b being movable by means of hydraulic pressure from a hydraulic device 14. The hydraulic pressure from the hydraulic device 14 is adjustable using a hydraulic control valve 15. Therefore, controlling the hydraulic control valve 15 enables control of the position of the movable sheave 12b in the direction of the shaft. The sheaves

12a, 12b each have a substantially conic shape, with the space between their opposing surfaces increasing outwardly. When the movable sheave 12b is caused to move closer to the fixed sheave 12a by the hydraulic pressure from the hydraulic device 14, the space between the sheaves 12a, 12b becomes narrower, thereby increasing the effective diameter of the pulley 12. When, on the other hand, the movable sheave 12b is caused to move away from the fixed sheave 12a by the hydraulic pressure from the hydraulic device 14, the space between the sheaves 12a, 12b becomes larger, thereby increasing the diameter of the driving pulley 12.

[0022]

A belt 16 is wound around the driving pulley 12, thereby connecting the following pulley 18 with the driving pulley 12. The belt 16 comprises a number of layers of flat-shaped blocks, which are tightened by a hoop.

[0023]

The following pulley 18 has a structure similar to that of the driving pulley 12, wherein substantially conic fixed sheave 18a and movable sheave 18b are opposed, the movable sheave 18b being movable by a hydraulic device 20. Also in the following pulley 18, the effective diameter of the following pulley 18 becomes larger as the movable sheave 18b moves closer to the fixed sheave 18a and becomes smaller as the movable sheave 18b moves away from the fixed sheave 18a. The following pulley 18 is connected to an output shaft 22 to transmit power to a vehicle wheel.

[0024]

By controlling the hydraulic pressure applied to the driving and following pulleys 12, 18 to determine the effective diameters of the driving and following pulleys 12, 18, the speed changing ratio is controlled. In the present embodiment, hydraulic pressure control to determine a speed changing ratio is applied to the driving pulley 12, while hydraulic pressure control to achieve power transmission with optimum transmission efficiency is applied to the following pulley 18. The force generated by the hydraulic pressure is referred to as the pulley thrust, which is a force acting in the direction of the shafts of the driving and following pulleys 12, 18, which together clamp the belt 16, and clamping the belt 16. That is, controlling the components to achieve appropriate pulley thrust of the driving and following pulleys 12, 18 realizes a speed changing ratio as commanded and maintains appropriate power transmission efficiency while preventing slip of the belt 16.

[0025]

Next, a structure for such control will be described. First, based on vehicle information including a vehicle speed, an input of the accelerator, and so forth, a speed ratio command value determination section 30 determines a speed ratio command value corresponding to a speed changing ratio, the speed ratio command value being a rotation speed ratio between the driving and following pulleys 12, 18. The determined speed ratio command value is supplied to a driver-side hydraulic pressure command value determination section 32. Meanwhile, the rate of rotation

of the input shaft 10, determined by a driver-side rotation rate determination section 34, and the rate of rotation of the output shaft 22, determined by a follower-side rotation rate determination section 36, are both supplied to a speed ratio calculation section 38, where a speed ratio between the input and output shafts 10, 22 is calculated, and the calculated speed ratio is supplied to the driver-side hydraulic pressure command value determination section 32.

[0026]

The driver-side hydraulic pressure command value determination section 32 compares the speed ratio command value supplied from the speed ratio command value determination section 30 and an actual speed ratio supplied from the speed ratio calculation section 38 and determines a driver-side hydraulic pressure command value. Here, because increasing the hydraulic pressure can enlarge the effective diameter of the driving pulley 12, thereby increasing a speed changing ratio, the hydraulic pressure command value is determined such that the speed ratio is set as commanded. Here, it should be noted that the speed ratio and the speed changing ratio have a one-to-one relationship, and either term will be used as appropriate.

[0027]

The determined hydraulic pressure command value is supplied to a driver-side hydraulic pressure command value adjustment section 40, to which a determined hydraulic pressure value is also supplied from a driver-side hydraulic pressure determination section 42, which determines a driver-side

hydraulic pressure, that is, an output hydraulic pressure from the hydraulic device 14. The driver-side hydraulic pressure command value adjustment section controls a driver-side hydraulic control valve 15 based on the hydraulic pressure command value and the determined hydraulic pressure value so as to feedback control the hydraulic pressure of the hydraulic device 14.

[0028]

Also, determined values by the driver-side rotation rate determination determination section 34 and the driver-side hydraulic pressure determination section 42 are supplied to a driving pulley thrust calculation section 44. The driver-side rotation rate determination section 34 calculates the force in the direction of the shaft of the pulley 12 based on the hydraulic pressure and centrifugal force based on the rate of rotation, and also calculates driving pulley thrust, or a clamping force of the driving pulley 12 which acts on the belt 16.

[0029]

Meanwhile, the hydraulic pressure of a follower-side hydraulic device 20 is determined by a follower-side hydraulic pressure determination section 46 and supplied to a following pulley thrust calculation section 48. The following pulley thrust calculation section 48, to which a value determined by the follower-side rotation rate determination section 36 is also supplied, calculates thrust of the following pulley based on these determined values.

[0030]

Then, the thrust of the driving pulley 12, calculated by the driving pulley thrust calculation section 44, and the thrust of the following pulley 18, calculated by the following pulley thrust calculation section 48, are supplied to a thrust ratio calculation section 50, where a thrust ratio is calculated by dividing the driving pulley thrust by the follower-side thrust.

[0031]

The thrust ratio calculated by the thrust ratio calculation section 50 is supplied to a thrust ratio state of change identifying section 52. The thrust ratio state of change identifying section 52, to which the thrust of the following pulley 18 is also supplied from the following pulley thrust calculation section 48, identifies the state of change of the thrust ratio according to changes in the thrust based on the supplied values.

[0032]

The output from the thrust ratio state of change identifying section 52 is supplied to a follower-side hydraulic pressure command value determination section 54. Based on the supplied state of change of the thrust ratio, the follower-side hydraulic pressure command value determination section determines a point where the direction of changing of a thrust ratio is inverted (a point where a thrust ratio peaks) according to changing of the thrust, and then determines a hydraulic pressure command value so as to control the thrust of the following pulley 18 such that the thrust ratio approaches that point. To the determined hydraulic pressure command value, a low frequency

excitement signal from a hydraulic pressure exciting section 56 is added, and the resultant value is supplied to a follower-side hydraulic pressure command value adjustment section 58. That is, the excitement signal causes the follower-side hydraulic pressure command value to change periodically around a target value.

[0033]

The follower-side hydraulic pressure command value adjustment section 58, which is also supplied with a value determined by the follower-side hydraulic pressure determination section 46, applies feedback control to a follower-side hydraulic pressure control valve 60 so as to cause the hydraulic device 20 to generate hydraulic pressure as commanded.

[0034]

As described above, in the present embodiment, the driving pulley 12 is used to control the thrust of the driving pulley 12 so that the speed ratio (a speed changing ratio) between the driver and follower sides assumes a value as commanded. On the other hand, on the follower side, based on the state of change of the thrust ratio as a ratio between the following pulley thrust and the driver-side thrust which is associated with the change in the follower-side thrust, the following pulley thrust is controlled so as to approach a point where the thrust ratio changes (i.e., peaks).

[0035]

Here, thrust control based on the state of change of a

thrust ratio will be described. Fig. 2 shows thrust ratios and changing rates of an active arc portion on the driving and following pulleys with respect to changing thrust of the following pulley under conditions of a constant speed ratio (1 or greater) and input torque. An active arc portion refers to a portion that contributes to power transmission in a pulley.

[0036]

An experiment for determining the active arc portion and respective pulley thrust was carried out in which following pulley thrust was initially set sufficiently large as shown in the right half of the drawing and gradually reduced. As the following pulley thrust decreases, the active arc portion gradually increases and the thrust ratio also increases to the point (its peak) indicated by the broken line in the drawing, after which it begins to decrease.

[0037]

Figs. 3 to 8 illustrate the state of transmission by the belt (a block pressing force) and hoop tension depending on the position of the belt 16. In the belt position between A and B, the belt 16 winds around the driving pulley 12 and does not contribute to a moving force of the belt 16 (a block pressing force). The belt position between B and C corresponds to an active arc portion on the driving pulley side. In the belt position between D and E, the belt 16 winds around the following pulley. The belt position between E and F corresponds to an active arc portion on the follower side. The area P1+P2, corresponding to an area corresponding to a hoop tension on the

driving pulley deducted by an area corresponding to a block pressing force in the active arc portion, corresponds to the thrust which acts on the driving pulley (a driving pulley thrust). The area S1+S2, corresponding to an area corresponding to a hoop tension on the following pulley deducted by an area corresponding to a block pressing force in the active arc portion, corresponds to the thrust acting on the following pulley (a following pulley thrust). With respect to the hoop tension of the pulleys, P1, S1 represent areas where the hoop tension is larger than the block pressing force, while P2, S2 represent areas where the hoop tension is smaller than the block pressing force. It should be noted that the areas of hoop tension shown above the range of active arc portions correspond to a force required to be applied to the belt 16 corresponding to a transmission torque (a block pressing force).

[0038]

Figs. 3 and 4 show a state where the following pulley 18 has sufficiently large thrust and excess thrust is thus available. In this state, because the hoop tension is sufficiently large throughout the entire region, a necessary block pressing force can be obtained despite a small active arc portion.

[0039]

Figs. 5 and 6 show a state where the thrust is reduced from the state of Figs. 3 and 4. In this case, the active arc portion area changes only slightly, while the thrust P1, S1 is reduced remarkably. Although, with respect to the degree of reduction, the reduction of P1, ΔP1, is larger than the reduction

of S1, $\Delta S1$, the area P2 is sufficiently larger than the area S2 (P2>>S2). Therefore, the thrust ratio (P1+P2)/(S1+S2) increases.

[0040]

Figs. 7 and 8 show a state where the thrust is further reduced from the state of Figs. 5 and 6. In this state, the active arc portion increases remarkably and a decrease of P2 and an increase of S2 are notable. Therefore, the thrust ratio (P1+P2)/(S1+S2) decreases.

[0041]

As described above, when the rate of change of the active arc portion increases, the increasing thrust ratio begins to decrease. This occurs just before the belt 16 begins to experience large slip (macro-slip), or near the point of maximum transmission efficiency in Fig. 22.

[0042]

It has been confirmed that this phenomenon occurs when the speed ratio is 1 or less, or even when the excess thrust is decreasing while the thrust remains constant and the input torque increases.

[0043]

Fig. 9 shows characteristics of thrust ratios and transmission efficiency according to following pulley thrust (secondary thrust) for various speed ratios. As shown, while the following pulley thrust is decreasing, large slip (macro-slip) begins to occur, causing the transmission efficiency to drop sharply. However, the thrust ratio peaks

immediately before this sharp drop. That is, although the thrust ratio peaks just before the transmission efficiency reaches maximum, the efficiency is at a sufficiently high level. Moreover, for a larger speed ratio, the thrust ratio peaks at a point closer to the point of the maximum transmission efficiency, although the thrust ratio peaks well before the point of maximum transmission efficiency for a smaller speed ratio. Further, the larger the speed ratio, the larger the increase of transmission efficiency due to reduction of the thrust. Therefore, larger improvement in the transmission efficiency through thrust control, such that the thrust ratio peaks, is expected for a larger speed ratio. Therefore, it is understood that control according to the present embodiment produces a larger effect during high speed operation.

[0044]

This phenomenon can be explained using Euler Theory, and Fig. 10 is a diagram illustrating this phenomenon based on Euler Theory. It can be seen from the drawing that the thrust ratio peaks at a position where the active arc portion begins to increase sharply. Therefore, it is understood that controlling a pulley thrust (secondary thrust) such that the thrust ratio approaches its peak can realize control of the thrust which achieves highly efficient power transmission while preventing macro-slip from taking place. When the active arc portion reaches 100%, large slip (macro-slip) begins to occur. Thus, it is important to maintain the pulley thrust (secondary thrust) higher than this point.

[0045]

In the present embodiment, thrust of the following pulley 18 is changed by the hydraulic pressure exciting section 56, and the resulting state of change of the thrust ratio is observed. A point at which the state of change switches between increasing and decreasing (a thrust ratio peak) is detected, and the thrust of the following pulley is controlled such that the thrust ratio approaches this point. This control makes it possible to maintain substantially maximum power transmission efficiency while preventing macro-slip of the belt 16 from taking place.

[0046]

In the following, specific examples of methods for determining a thrust control value based on the state of change of the thrust ratio (a thrust ratio peak) are described.

[0047]

(i) A method based on detection of phase change

This method utilizes a secondary or higher model for estimation of pulley thrust for thrust control and of the phase of a thrust ratio within a range of ± 180°. A model parameter is estimated using a successive least squares method. It should be noted that a linear model can estimate a phase only within a range of ± 90°.

[0048]

First, values for change of thrust ratio when a sinusoidal wave is input to pulley thrust are input to an identifying model (secondary), and a model parameter is estimated using a successive least squares method. Using the estimated model

parameter, the phase of the identifying model at a predetermined frequency is estimated.

[0049]

It should be noted that a point at which the estimated phase (a phase delay) changes by a predetermined amount or greater or reaches a predetermined value is determined as the peak of a thrust ratio, and that a region with an advancing phase than the peak is determined to be a region with the same phase, while a region with a lagging phase is determined to be a region with an opposite phase. A region with an opposite phase has excess thrust, while a region with the same phase does not.

[0050]

When the estimated phase of the identifying model relative to the excitement frequency is the same phase, the pulley thrust (following pulley thrust: secondary thrust) may be controlled so as to reduce the thrust. When the phase is opposite, on the other hand, the pulley thrust may be controlled so as to increase the thrust.

[0051]

(ii) A method based on detection of gain change

A secondary or higher model is used as in the above-described method (i), to which a pulley thrust and a thrust ratio are input, and a model parameter is estimated using a successive least squares method. Then, a gain of the identifying model at a predetermined frequency is obtained. A point at which the gain of the model changes from decreasing to increasing while the pulley thrust is decreasing is determined to be the peak

of a thrust ratio.

[0052]

That is, a region where the gain decreases or remains unchanged while the pulley thrust is decreasing has excess thrust, while the excess thrust is decreasing in a region where the gain increases while the pulley thrust is decreasing.

[0053]

(iii) A method using phase and gain

A secondary or higher model is used as in the above-described method (i), and a model parameter is estimated using a successive least squares method. Then, the peak of a thrust ratio is obtained using both the phase and the gain of the identifying model at a predetermined frequency. That is, the peak of thrust is obtained according to the results of checks (i) and (ii). Thus, this method enables more preferable control.

[0054]

(iv) A method based on detection of gradient 0

Change of thrust while pulley thrust is decreasing is detected so that a point at which the gradient of the thrust ratio becomes 0 is determined as the peak of the thrust ratio. A region wherein a gradient of the thrust ratio increases while pulley thrust decreases is determined to be a region having excess thrust, while excess thrust is decreasing in a region where a gradient of the thrust ratio decreases while pulley thrust decreases.

[0055]

(v) A method based on detection of maximum thrust ratio

This method is basically similar to the above-described method (iv) except that change of thrust while pulley thrust is decreasing is observed so that the maximum value is determined.

[0056]

Among these, the method (i) is most practical, which will be described with reference to Fig. 11. Following pulley thrust and the calculated thrust ratio are input to a successive least squares identifying section 52a within the thrust ratio state of change identifying section 52, wherein a model parameter of a secondary or higher identifying model is estimated using the least squares method. The estimated model parameter is input to a phase calculation section 52b, where a phase at a predetermined frequency is calculated using the estimated model parameter, the predetermined frequency corresponding to an excitement frequency.

[0057]

It should be noted that the successive least squares method is not described here because it is a well-known method as described in, for example, "System Control Information Library 9, System Identification Introduction", pp. 71-86, Asakura Shoten (1994/5).

[0058]

The obtained estimated phase is input to a thrust control amount map 54a of the follower-side hydraulic pressure command value determination section 54. The thrust control amount map 54a, which stores in advance thrust control amounts (a hydraulic pressure) relative to phases, outputs a corresponding control

amount in response to an input of an estimated phase. Then, the output control amount is input to an adder 54b, wherein the control amount is added to a thrust command value in the last (one-previous) cycle to thereby obtain thrust command value (a hydraulic pressure command value).

[0059]

As described above, preparation of the thrust control amount map 54a allows determination of an appropriate hydraulic control amount relative to a concerned phase. In addition to the method using a thrust control amount map 54a, a hydraulic control amount may alternatively be determined using a feedback control such as a PID control so as to maintain a target phase.

[0060]

In the following, change of a thrust ratio caused when the following pulley thrust (hydraulic pressure) is excited by a sinusoidal wave will be described with reference to Figs. 12, 13.

[0061]

Values for thrust ratios relative to following pulley thrust are as shown in Fig. 12. As the thrust decreases, the thrust ratio gradually increases and, after it overshoots the peak, sharply drops.

[0062]

In the region on the right side of the peak, where excess thrust is available, a thrust ratio output (A) in response to a sinusoidal wave input (A) has a small gain and an opposite phase as shown in the drawing. On the other hand, after it

overshoots the peak, a thrust ratio output (B) in response to an input (B) has a large gain and the same phase as shown in the drawing. Thus, a method such as the above-described methods (i) to (v) is used to detect this change.

[0063]

Here, Fig. 13 shows change of a gain (dB) and phase (dB) relative to an excitement frequency (a secondary hydraulic excitement frequency) applied to a following pulley thrust. It can be seen from the drawing that, relative to excitement frequencies of about 1 to 10 Hz, the gain and phase when the thrust ratio has passed its peak and excess thrust is thus not available differ from the gain and phase of other cases where the thrust ratio is yet to pass the peak, and are therefore distinguishable. In particular, it can be seen that, when change of phase is used, the peak of a thrust ratio can be readily determined relative to excitement frequencies of about 1 to 10 Hz.

[0064]

Fig. 14 shows results of an experiment in which the following pulley thrust was actually controlled such that the thrust ratio peaked. Beginning of control triggers phase estimation. Because sufficiently high following pulley thrust was ensured at this time, an opposite phase then resulted. On the other hand, as the control began, the hydraulic pressure began decreasing and transmission efficiency improved. It can be confirmed that controlling the phase of the thrust ratio to be -90° (a predetermined phase delay), that is, the boundary

between the same phase and an opposite phase, can realize a hydraulic pressure having an appropriate value and improve the transmission efficiency.

[0065]

Alternatively, the hydraulic pressure exciting section 56 may be removed so as not to intentionally excite the hydraulic pressure (thrust). In other words, during actual control, the hydraulic pressure fluctuates at various frequencies, even when no active exciting section is provided. Then, detection of a response with respect to a preferable frequency of the order of a few Hz (for example, 2 Hz) among those frequencies can realize processing similar to the above.

[0066]

Fig. 15 shows a result of control performed using a structure without a hydraulic pressure exciting section 56. As shown, it is possible to control the thrust of the following pulley 18 such that the thrust ratio is maintained at its peak even through the hydraulic pressure is not actively excited.

[0067]

"Examples of Other Structures"

Fig. 16 shows an example of a structure for controlling a speed ratio using the following pulley. In the example of Fig. 16, the speed ratio is controlled using the following pulley 12 so that the thrust of the driving pulley 12 is controlled using the driving pulley 12 such that the thrust ratio is maintained at its peak.

[0068]

For this purpose, the follower-side hydraulic pressure command value determination section 54 determines a follower-side hydraulic pressure command value based on signals from the speed ratio command value determination section 30 and the speed ratio calculation section 38. Meanwhile, the thrust ratio state of change identifying section 52 identifies the state of change of the thrust ratio with respect to change of the thrust on the driver-side 12, based on the thrust ratio supplied from the thrust ratio calculation section 50 and the driving pulley thrust supplied from the driving pulley thrust calculation section 44. Then, based on the identification result, the driver-side hydraulic pressure command value determination section 32 determines a driver-side hydraulic pressure. Further, an excitement signal from the hydraulic pressure exciting section 56 is added to the driver-side hydraulic pressure command value, whereby the driver-side hydraulic pressure is excited.

[0069]

As described above, in this embodiment, a driver-side hydraulic pressure is controlled to thereby control driver-side thrust such that the thrust ratio approaches close to its peak. The effects and advantages similar to those of the above embodiment can thereby be achieved.

[0070]

It should be noted that, as described above, either of the driving or following pulley 12, 18 can be desirably selected for use in determining a speed ratio for thrust control. This

arrangement shown in Fig. 16 can be introduced to any of the following embodiments.

[0071]

Fig. 17 shows an embodiment in which a hydraulic pressure command value is used in thrust estimation. In this embodiment, the follower-side hydraulic pressure determination section 46 and the driver-side hydraulic pressure determination section 42 are omitted. Because a determined hydraulic pressure value is not available and feedback control based on the determined value is therefore not possible, the driver-side hydraulic pressure command value adjustment section 40 and the follower-side hydraulic pressure command value adjustment section 58 are also eliminated.

[0072]

Instead, a hydraulic pressure command value supplied to the follower-side hydraulic pressure control valve 60 is also supplied to the following pulley thrust calculation section 48, while a hydraulic pressure command value supplied to the driver-side hydraulic control valve 15 is also supplied to the driving pulley thrust calculation section 44.

[0073]

Fig. 18 shows relationships between hydraulic pressure command values and thrust ratios. As shown, gradual decrease in hydraulic pressure command value causes a thrust ratio to change. That is, it is understood that a hydraulic pressure command value can be handled substantially equivalent to a determined hydraulic pressure value. It should be noted that

the hydraulic pressure command value is subjected to low-pass filtering whereby high frequency components are removed.

[0074]

As described above, the use of a hydraulic pressure command value instead of a value of a hydraulic pressure can provide similar effects and advantages.

[0075]

Fig. 19 shows an example of a structure for use when the rotation fluctuation can be assumed to be small. In this structure, supply to the driving pulley thrust calculation section 44 of the rate of rotation supplied form the driver-side rotation rate determination section 34 is omitted, and supply to the following pulley thrust calculation section 48 of the number of rotations supplied from the follower-side rotation rate determination section 36 is also omitted. Thus, driving pulley thrust calculation section 44 and following pulley thrust calculation section 48 calculate pulley thrusts without consideration of the rate of rotation. This is not problematic because the influence of the rate of rotation is small. This method can remarkably reduce a computation load and is preferable for use during low speed operation.

[0076]

Fig. 20 shows an example of a structure for controlling a thrust ratio to its peak through the use of fluctuation of a driving torque. In this example, a driving torque, which is transmitted through the input shaft 10, is determined by a driving torque determination section 70, and a driving torque exciting

section 72 excites the driving torque in the order of a few Hz.

[0077]

The thrust ratio state of change identifying section 52 calculates a suitable pulley thrust based on the state of change of a thrust ratio relative to changing of a driving torque. That is, whereas in the above example the relationship between a pulley thrust and a thrust ratio is determined based on the assumption that a driving torque is constant, providing a predetermined fluctuation to the driving torque to measure response of the thrust ratio relative to that fluctuation can be equivalent to fluctuating pulley thrust to ascertain change of the thrust ratio. That is, increasing a driving torque is equivalent to reducing pulley thrust. Then, pulley thrust is controlled based on the state of change of a thrust ratio caused by increasing the driving torque, whereby a peak thrust ratio can be maintained. Here, because the relationship between the phase of a driving torque, or an input, and the phase of a thrust ratio, or an output, is opposite from that of Fig. 1, when the phase of excitement applied to the driving torque and the phase of the thrust ratio, or an output, are of the same phase, it can be understood that an excess thrust exists and the thrust should be reduced. On the other hand, when the phases are opposite, it can be understood that thrust is insufficient and that the thrust should be increased.

[0078]

Also in this case, intentional excitement of the driving torque is not required, and the driving torque exciting section 72 can be omitted.

[0079]

As an alternative to the example of Fig. 20 in which the driving torque is caused to fluctuate, the pulley thrust can be controlled based on change of a thrust ratio due to ground surface disturbance.

[0080]

Specifically, when a load torque acting on a tire due to disturbance from the ground is determined and change of a thrust ratio relative to the determined load torque is determined, a pulley thrust can be controlled based on the relationship between changing of the thrust ratio and the thrust ratio. This method is basically the same as a method in which a driving torque is made fluctuating.

[0081]

When the rate of rotation of a tire is reduced due to ground disturbance, the rate of rotation of the following pulley is also reduced, which then reduces centrifugal hydraulic pressure. The reduction of the rate of rotation corresponds to reduction of pulley thrust. Then, based on the relationship between change of the rate of rotation of a tire or following pulley and change of the thrust ratio, the pulley thrust may be controlled so that the thrust ratio can be maintained at a predetermined value. Here, it should be noted that, in a driving pulley, the driving pulley thrust is controlled to control the speed ratio.

[0082]

In any of the above examples, thrust ratio is controlled to maintain its peak. Instead of the thrust ratio, a ratio of

average friction coefficients can also be employed to thereby perform thrust optimization control.

[0083]

Respective variables are defined as follows: Ti = an input torque; μp = an average friction coefficient between a driving pulley and a belt; Fp = thrust of the driving pulley; Rp = a belt hanging diameter in the driving pulley; Ip = rotational inertia of the driving pulley; dNp = rotational acceleration of the driving pulley; T = torque transmitted by the belt; μs = an average friction coefficient between a following pulley and the belt; Fs = thrust of the following pulley; and Rs = a belt hanging diameter in the following pulley.

[0084]

In this case,

[Expression 1]

$$Ti=Ip\cdot dNp+\mu p\cdot Fp\cdot Rp=Ip\cdot dNp+T$$

$$T=\mu s\cdot Fs\cdot Rs$$

$$\mu p=(Ti-Ip\cdot dNp)/(Fp\cdot Rp)$$

$$\mu s=(Ti-Ip\cdot dNp)/(Fs\cdot Rs)$$

[0085]

The average friction coefficient ratio will be

[Expression 2]

$$\mu s/\mu p=Fp\cdot Rp/Fs\cdot Rs=(Fp/Fs)\cdot(Rp/Rs)$$

[0086]

Because the ratio between hanging diameters, or Rp/Rs,

is constant when a constant speed changing ratio is assumed, the thrust ratio Fp/Fs is proportional to the average friction coefficient ratio $\mu s/\mu p$. Therefore, an average friction coefficient ratio can replace the thrust ratio.

[0087]

Thus, the use of the ratio of average friction coefficients instead of a thrust ratio can also realize control to achieve optimum pulley thrust as described above. In particular, the use of the ratio of average friction coefficients can cancel changing of the thrust ratio caused when the speed changing ratio is changed. That is, when the above-described ratio of hanging diameters is considered, the ratio of average friction coefficients should be referred to regardless of the value of the speed changing ratio.

[0088]

Fig. 21 shows a structure for controlling pulley thrust based on the ratio of averaged friction coefficients. A belt hanging diameter determination section 80 determines belt hanging diameters of the driving pulley 12 and the following pulley 18, respectively.

[0089]

Specifically, the belt hanging diameter determination section 80 may determine the position of the top of the belt block as a belt hanging diameter, which can be measured using a non-contact displacement measurement device of an optical or magnetic type. Alternatively, because the distance between sheaves is determined by the position in the direction of the

shaft of the pulley and the belt hanging diameter can be determined based on that distance, the position in the direction of the shaft of the pulley may be measured. Still alternatively, calculations may be based on a speed changing ratio.

[0090]

The value determined by the belt hanging diameter determination section is supplied to an average friction coefficient ratio calculation section 82. The friction coefficient ratio calculation section 82, to which a thrust ratio from the thrust ratio calculation section 50 is also supplied, replaces the thrust ratio with the average friction coefficient ratio based on the above expression described in Expression 2. The resulting average friction coefficient ratio is supplied to an average friction coefficient ratio state of change identifying section 84, where the pulley thrust is controlled such that the average friction coefficient ratio is located near its peak. This estimation method can be performed in a manner similar to the above-described calculation of the peak of a thrust ratio. Then, data concerning the peak of the average friction coefficient ratio is supplied to the follower-side hydraulic pressure command value determination section 54, where a hydraulic pressure command value is determined.

[0091]

Thus, the use of an average friction coefficient ratio enables control so as to achieve optimum pulley thrust, even when the speed changing ratio varies as described above.

[0092]

Further, in the above-described embodiments, the peak of a thrust ratio or average friction coefficient ratio is determined so that pulley thrust is controlled such that the thrust ratio or average friction coefficient ratio peaks. However, the relationship between these may be stored in a map so that the optimum thrust can be directly output according to the various conditions which govern a thrust ratio. This map is preferably rewritten through learning, according to the peak of thrust ratio which is calculated based on actual running conditions. This guarantees a higher speed response and allows control of a pulley thrust ratio such that the thrust ratio or average friction coefficient ratio peaks, similar to a case wherein control is executed through computation.

[0093]

Fig. 23 shows a structure for pulley thrust control which is capable of amending the control map using a thrust ratio peak estimation method, in a control system wherein a hydraulic pressure command value for controlling a pulley thrust is given as a control map including arguments such as an engine rotation speed Ne, an engine torque Te, a speed changing ratio γ, and so forth. In this example, hydraulic pressure (primary hydraulic pressure) control for controlling a speed ratio (a speed changing ratio) is performed using the driving pulley 12, while hydraulic pressure (secondary hydraulic pressure) control for controlling pulley thrust is performed using the following pulley 18.

[0094]

A primary hydraulic pressure from the primary control system 100 which controls the primary hydraulic pressure according to the speed changing ratio (a speed ratio) is supplied to the driving pulley 12. Meanwhile, the secondary hydraulic pressure from the secondary hydraulic pressure control system 102 is supplied to the following pulley 18.

[0095]

The primary and secondary hydraulic pressures are then supplied to a thrust ratio peak estimation device 104, which determines the state of change of the thrust ratio based on these supplied hydraulic pressures, and estimates a secondary hydraulic pressure corresponding to the peak of the pulley thrust ratio. The estimated secondary hydraulic pressure command value corresponding to the peak of a thrust ratio is supplied to a switch 106.

[0096]

Meanwhile, an output from the thrust ratio peak estimation device 104 is multiplied by a safety rate (a number slightly greater than 1) in a safety ratio multiplier 108, and is then supplied to a control map (a secondary hydraulic pressure control map) 110. Using an engine rotation speed Ne, an engine torque Te, and a speed changing ratio γ as an argument, the control map 110 outputs a secondary hydraulic pressure command value which corresponds to the peak of the thrust ratio. Then, the control map is amended based on the relationship between the value (the secondary hydraulic pressure command value) supplied from the thrust ratio peak estimation device 104 and a secondary

hydraulic pressure command value to be output at that time. An output from the control map 110, or the secondary hydraulic pressure command value, is also supplied to the switch 106.

[0097]

The switch 106 selects a secondary hydraulic pressure command value from the thrust ratio peak estimation device 104 only during the period when the thrust ratio peak estimation device 104 is performing estimation and supplies a secondary hydraulic pressure command value from the control map 110 to a secondary hydraulic pressure control system 102 during other periods.

[0098]

For actual use in control in a vehicle, use of the control map 110 facilitates control of a secondary hydraulic pressure and, thus, this control system is generally employed during running. However, because of differences unique to each vehicle, a general control map cannot be employed without amendment. Thus, the peak of a thrust ratio is estimated from a predetermine test running and the control map 110 is amended based on the results of the test. The amended control map 110 is employed during subsequent operation of the vehicle to control the secondary hydraulic pressure. Moreover, because the characteristics of a vehicle may change over time, the estimation by the thrust ratio peak estimation device 104 may be periodically performed for updating and amending of the control map 110.

[0099]

Such amendment of the control map 110 will next be

described.

[0100]

First, as described above, during general operation, a value from the control map 110 is used as a command value (a secondary hydraulic pressure command value) for a hydraulic pressure which controls pulley thrust.

[0101]

Estimation is desirably performed using the thrust ratio peak estimation device 104, the procedure being identical to that performed when amending to account for the uniqueness of each vehicle.

[0102]

During learning (while a thrust peak is being estimated), the switch 106 selects a secondary hydraulic pressure command value from the thrust ratio peak estimation device 104. Then, while slowly changing the hydraulic pressure command value into, for example, a ramp wave shape so that the pulley thrust gradually drops, the state of change of the pulley thrust ratio is observed and the hydraulic pressure command value at the time when the thrust ratio peaks is recorded.

[0103]

The peak of the pulley thrust ratio may be obtained based on changes in a gradient of the pulley thrust ratio. Alternatively, a point at which the estimated phase reaches a predetermined value or greater may be used. Still alternatively, a point at which the estimated phase changes by a predetermined amount or greater may be used.

[0104]

Upon completion of the recording of the hydraulic pressure command value, the switch 106 resets the hydraulic pressure command value to a value from the secondary hydraulic pressure control map 110, and rewrites the value in the control map 110 to be referred to when the thrust ratio peaks (a value to be output from the control map 110) to a value obtained by multiplying the recorded control command value by a predetermined safety factor.

[0105]

As described above, the control map 110 can be rewritten based on the state at that moment and the control map 110 can be maintained as an appropriate map.

[0106]

[Advantages]

As described above, according to the present invention, pulley thrust is controlled based on the state of change of the thrust ratio. The thrust ratio peaks immediately before a large belt slip (macro-slip) occurs and power transmission efficiency is maximized also immediately before macro-slip occurs. Thus, appropriate control of the pulley thrust can be performed by controlling the pulley thrust in accordance with the state of change of the thrust ratio.

[0107]

Because the thrust ratio peaks immediately before macro-slip occurs and the maximum point of power transmission efficiency exists also immediately before macro-slip occurs,

appropriate control of the pulley thrust can be performed by controlling the pulley thrust so as to closely approach the point where the gradient of changing of the thrust ratio changes.

[0108]

Yet further, by periodically changing pulley thrust, the peak of a thrust ratio can be easily detected.

[0109]

Yet further, by measuring a hydraulic pressure which defines the thrust of the driving and following pulleys, pulley thrust can be easily measured.

[0110]

Yet further, by determining, based on a command value, a hydraulic pressure which defines the thrust of the driving and following pulleys, determination means such as a hydraulic sensor can be omitted.

[0111]

Preferably, an average friction coefficient ratio is used in place of the thrust ratio so that the pulley thrust is controlled based on the state of change of the average friction coefficient ratio, the average friction coefficient ratio being obtained by multiplying the thrust ratio by a ratio between belt hanging diameters of the driving pulley and the following pulley. Because the average friction coefficient ratio changes according to the speed ratio, suitable thrust control can be performed even though the speed changing ratio changes.

[Brief Description of the Drawings]

[Fig. 1]

A diagram showing the general system structure of a pulley thrust control device of a belt-type continuously variable transmission unit according to a preferred embodiment.

[Fig. 2]

A diagram showing the relationship of a thrust ratio and an active arc portion with respect to following pulley thrust.

[Fig. 3]

A diagram showing a block pressing force when there is excess thrust.

[Fig. 4]

A diagram showing a hoop tension and pulley thrust when there is excess thrust.

[Fig. 5]

A diagram showing a block pressing force when thrust drops.

[Fig. 6]

A diagram showing a hoop tension and pulley thrust when thrust drops.

[Fig. 7]

A diagram showing a block pressing force when thrust further decreases.

[Fig. 8]

A diagram showing a hoop tension and pulley thrust when thrust further decreases.

[Fig. 9]

A diagram showing the relationship of transmission efficiency and a thrust ratio with thrust.

[Fig. 10]

A diagram to explain the decrease based on Euler Theory.

[Fig. 11]

A diagram showing a structure for generating a thrust command value.

[Fig. 12]

A diagram showing characteristics of a thrust ratio.

[Fig. 13]

A diagram showing the relationship of a phase and a gain with a hydraulic pressure excitement frequency.

[Fig. 14]

A diagram showing the relationship of a hydraulic pressure, transmission efficiency, and a thrust ratio phase.

[Fig. 15]

A diagram showing the relationship of a thrust ratio phase and a hydraulic phase without excitation of the hydraulic pressure.

[Fig. 16]

A diagram showing a system structure in which a speed ratio is controlled on a follower side.

[Fig. 17]

A diagram showing a system structure in which thrust is estimated using a hydraulic command value.

[Fig. 18]

A diagram showing thrust ratio characteristics when a hydraulic command value is used.

[Fig. 19]

A diagram showing a system structure for cases where the

rate of rotation can be assumed small.

[Fig. 20]

A diagram showing a system structure for cases where a driving torque fluctuation is used.

[Fig. 21]

A diagram showing a system structure for cases where an average friction coefficient ratio is used.

[Fig. 22]

A diagram showing the relationship of a belt slip rate and transmission efficiency with a transmission torque.

[Fig. 23]

A diagram illustrating a structure for updating a control map 110.

[Explanation of Reference Numerals]

10 INPUT SHAFT

12 DRIVING PULLEY

16 BELT

18 FOLLOWING PULLEY

44 DRIVING PULLEY THRUST CALCULATION SECTION

48 FOLLOWING PULLEY THRUST CALCULATION SECTION

50 THRUST RATIO CALCULATION SECTION

52 THRUST RATIO STATE OF CHANGE IDENTIFYING SECTION

54 FOLLOWER-SIDE HYDRAULIC PRESSURE COMMAND VALUE DETERMINATION SECTION

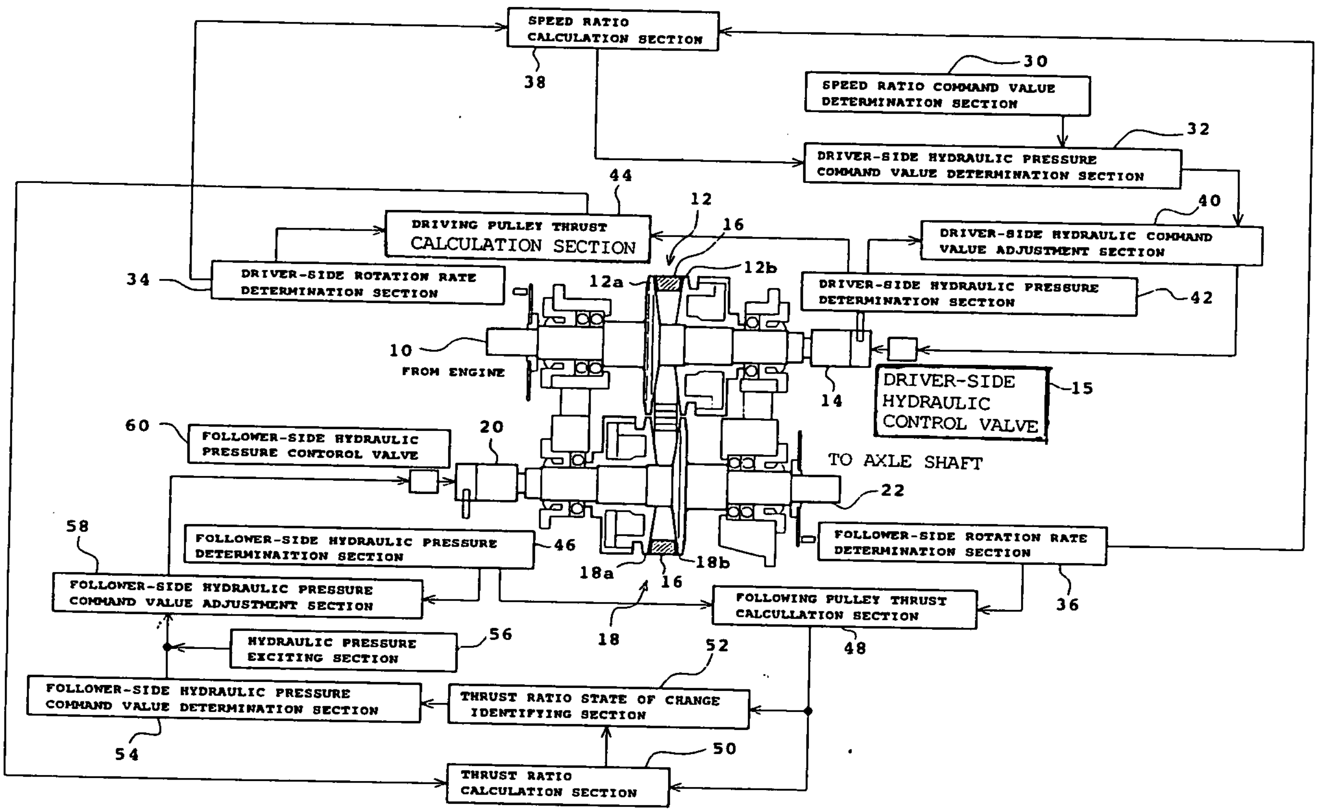
56 HYDRAULIC PRESSURE EXCITING SECTION

58 FOLLOWER-SIDE COMMAND VALUE ADJUSTMENT SECTION

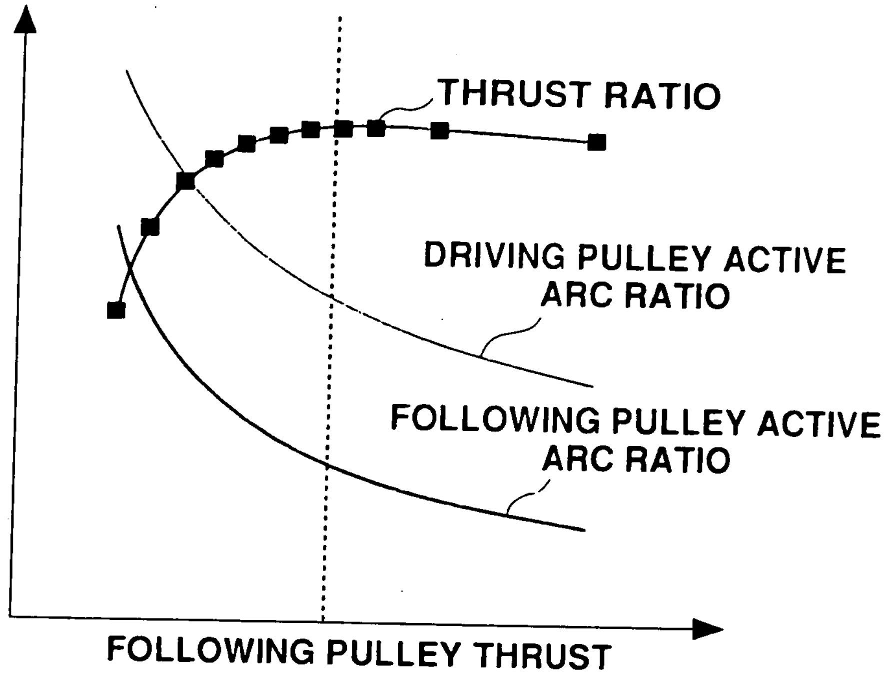
60 FOLLOWER-SIDE HYDRAULIC PRESSURE CONTROL VALVE

[Name of Document] Drawings

[Fig. 1]

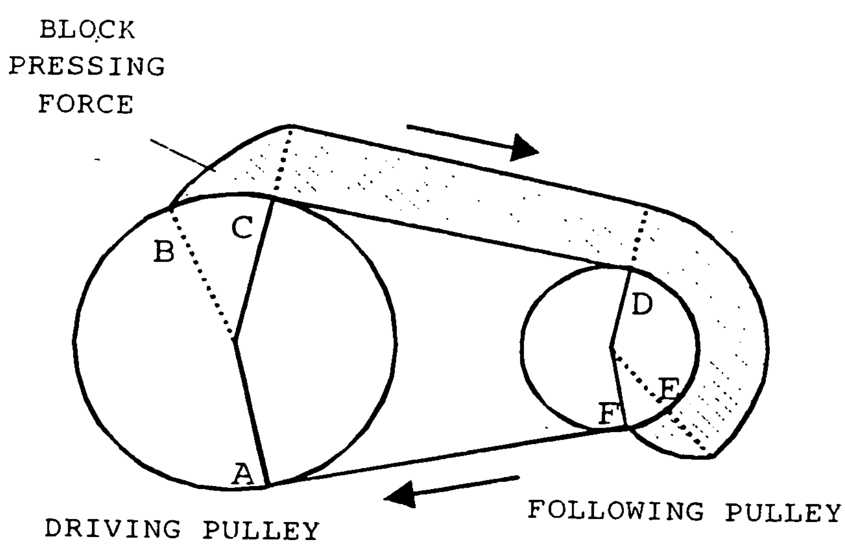


[Fig. 2]



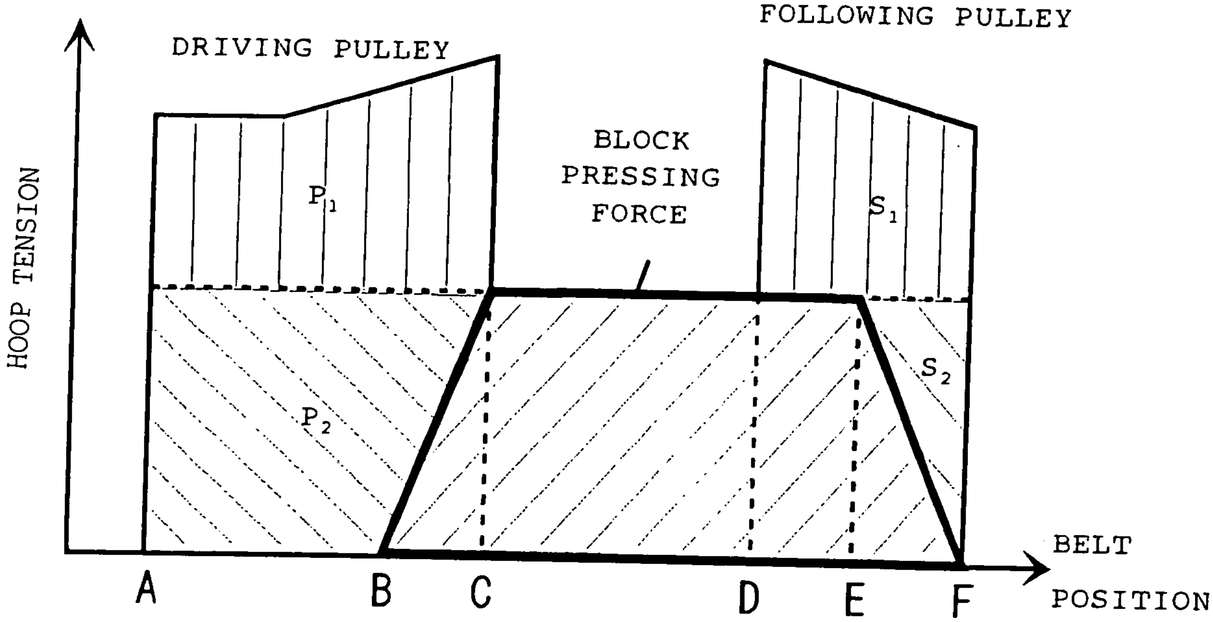
THRUST RATIO AND ACTIVE ARC RATIO

[Fig. 3]



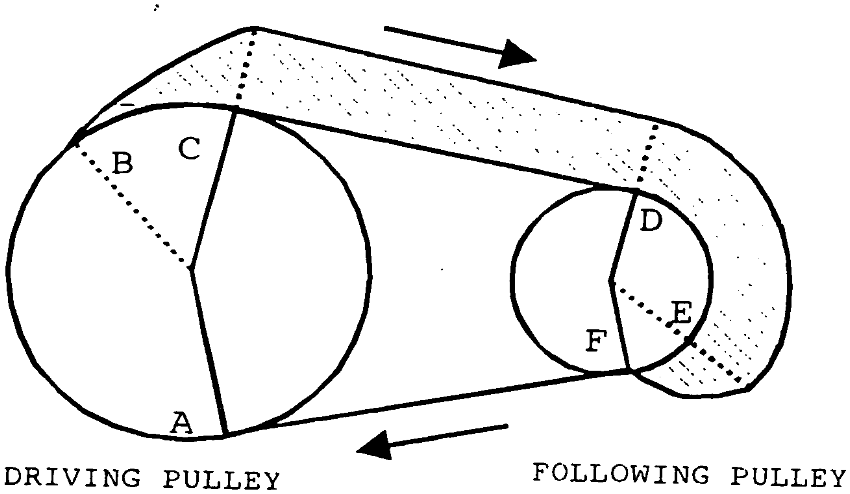
CASE WITH EXCESS THRUST

[Fig. 4]



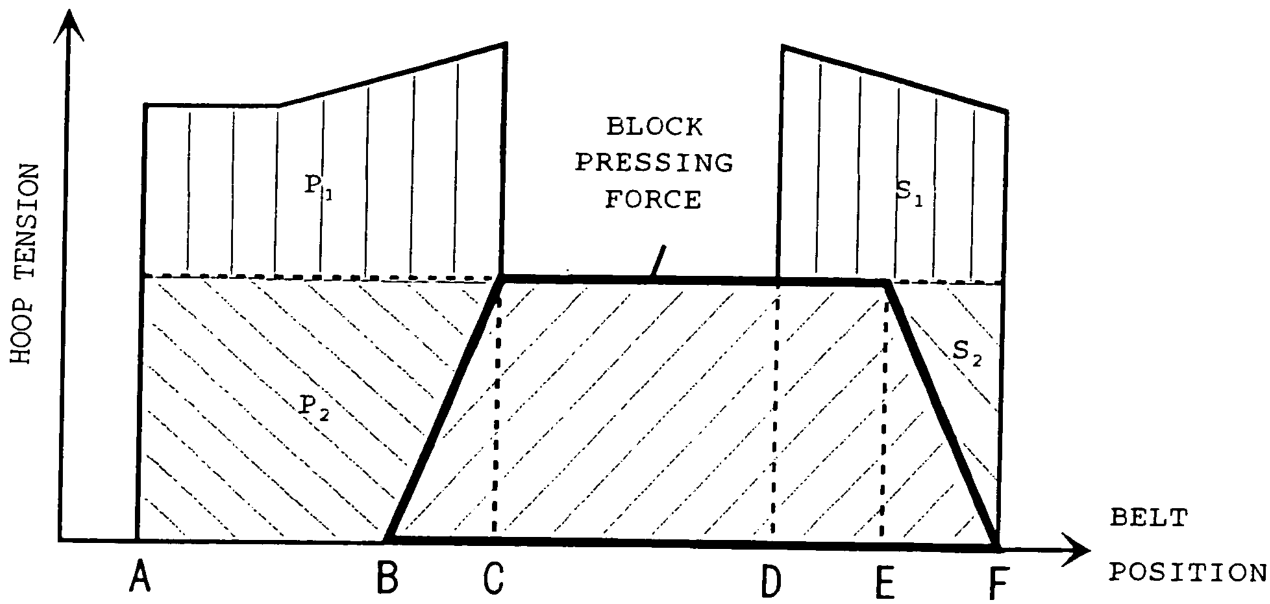
THRUST RATIO IN CASE WITH EXCESS THRUST

[Fig. 5]



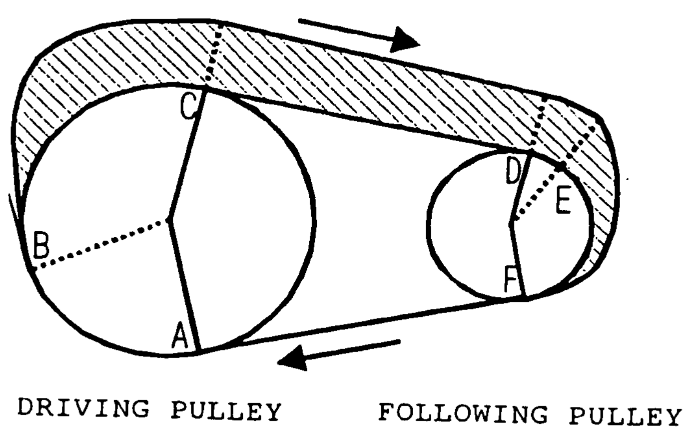
CASE WITH DECREASED THRUST

[Fig. 6]



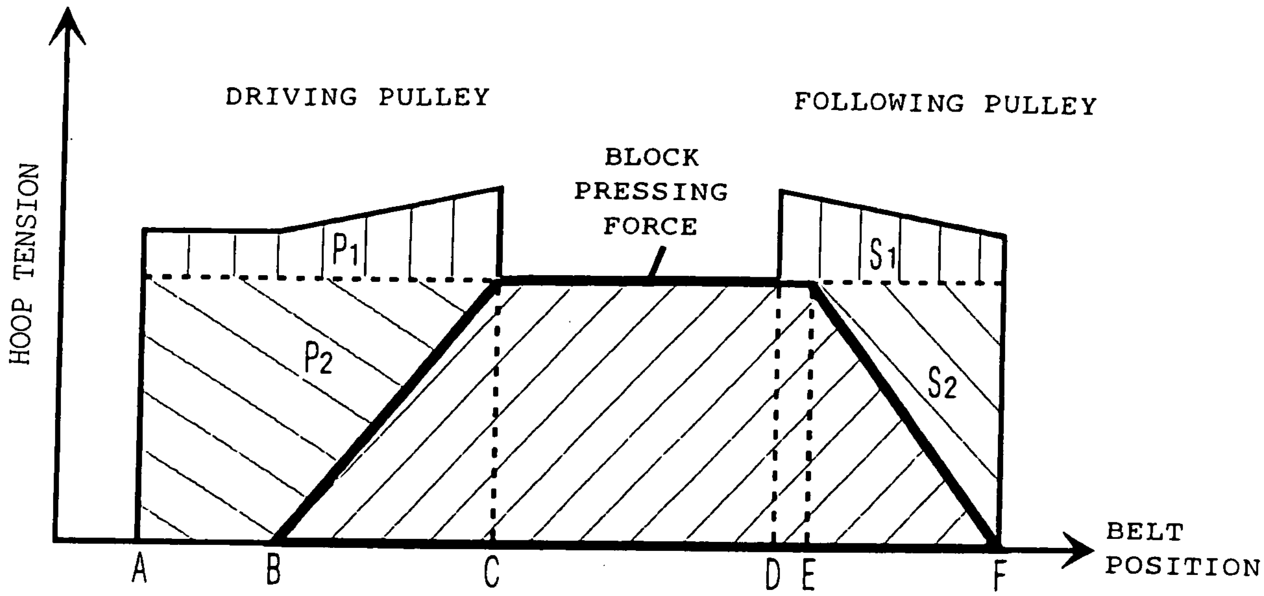
THRUST RATIO IN CASE WITH DECREASED THRUST

[Fig. 7]



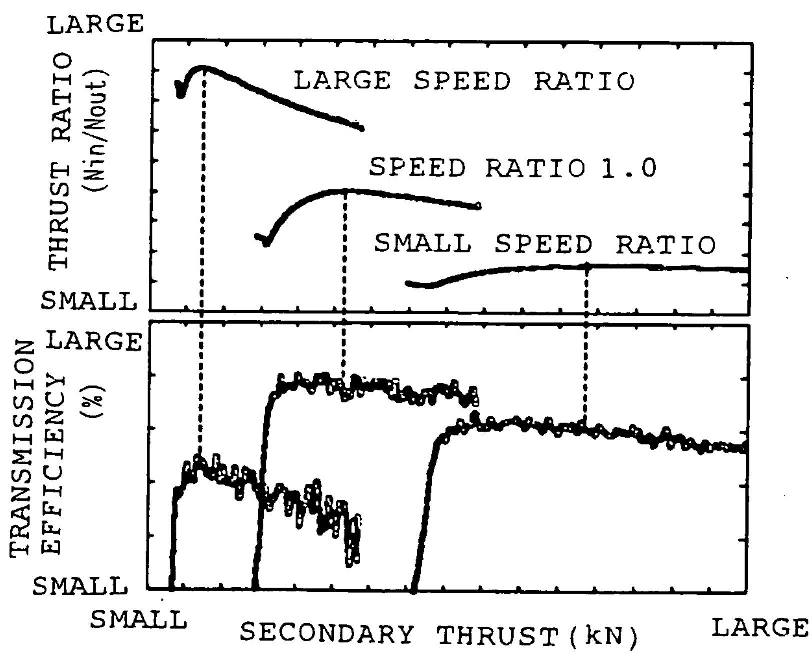
CASE WITH FURTHER DECREASED THRUST

[Fig. 8]



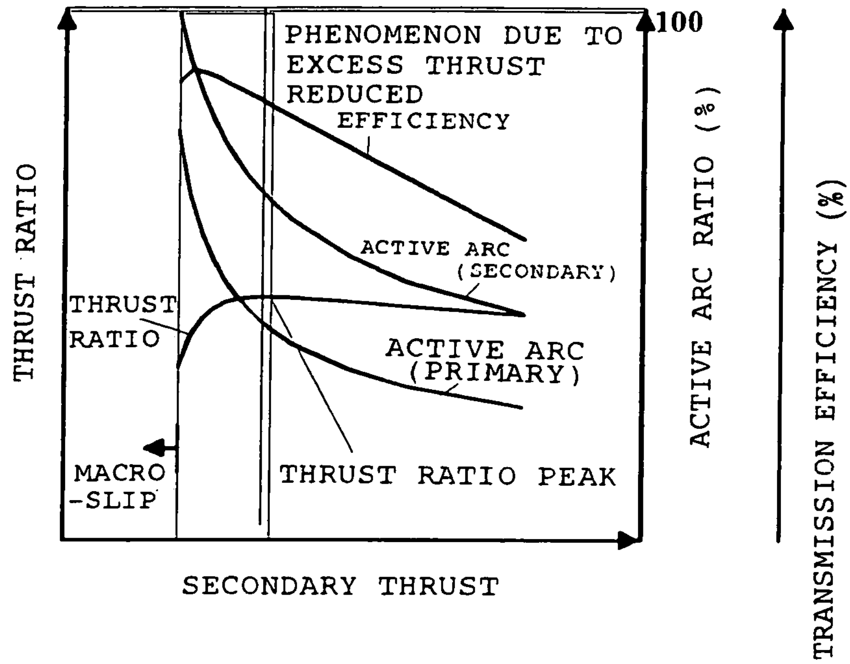
THRUST RATIO IN CASE WITH FURTHER DECREASED THRUST

[Fig. 9]

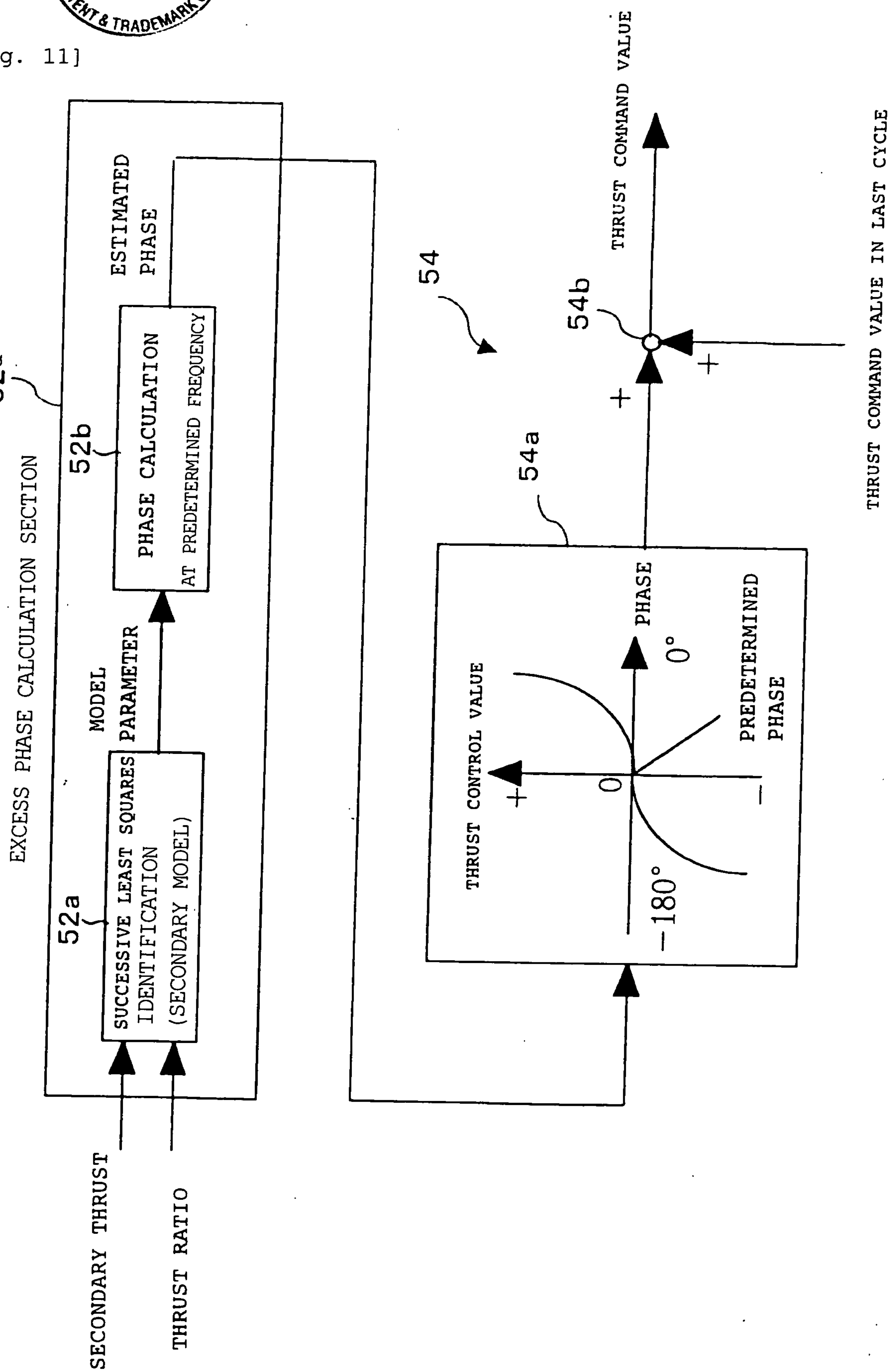


THRUST RATIO AND TRANSMISSION EFFICIENCY

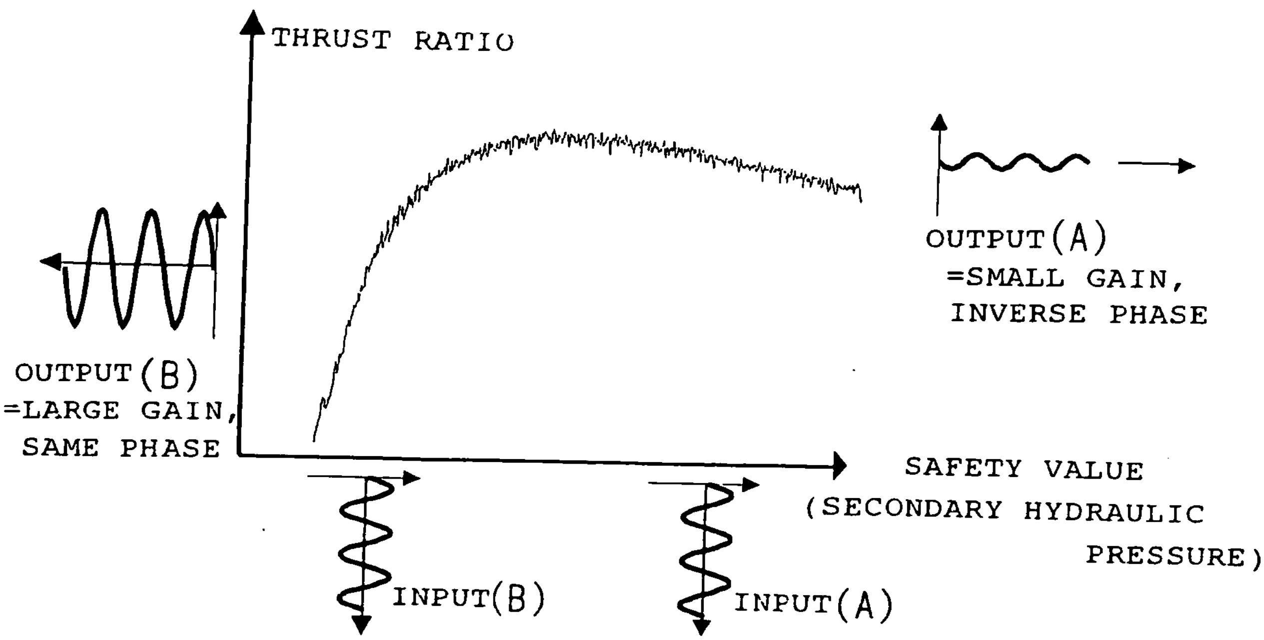
[Fig. 10]


PHENOMENON DUE TO EXCESS THRUST
REDUCED EFFICIENCY
100
THRUST RATIO
ACTIVE ARC RATIO (%)
TRANSMISSION EFFICIENCY (%)
ACTIVE ARC (SECONDARY)
THRUST RATIO
ACTIVE ARC (PRIMARY)
MACRO -SLIP
THRUST RATIO PEAK
SECONDARY THRUST

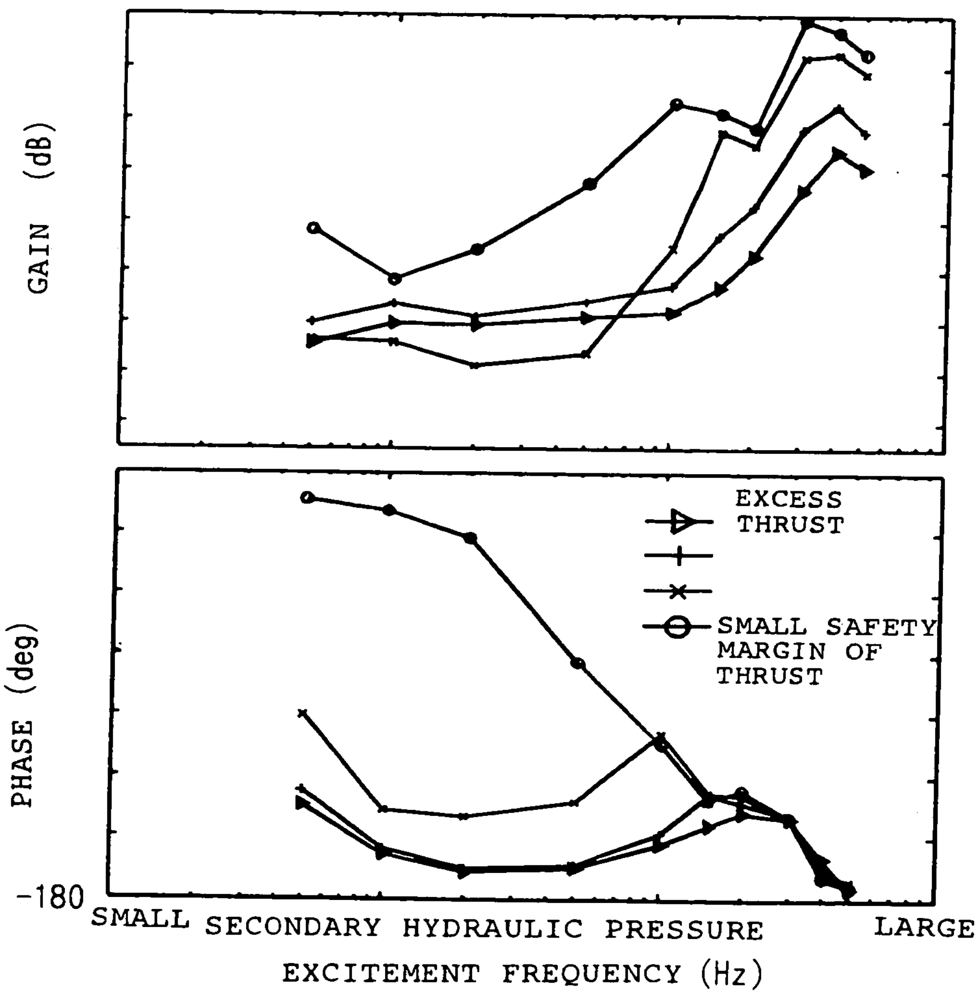
[Fig. 11]


EXCESS PHASE CALCULATION SECTION
52a
52a
SUCCESSIVE LEAST SQUARES IDENTIFICATION (SECONDARY MODEL)
MODEL PARAMETER
52b
PHASE CALCULATION AT PREDETERMINED FREQUENCY
ESTIMATED PHASE
SECONDARY THRUST
THRUST RATIO
54
54a
THRUST CONTROL VALUE
+
0
−
PHASE
0°
−180°
PREDETERMINED PHASE
54b
+
+
THRUST COMMAND VALUE
THRUST COMMAND VALUE IN LAST CYCLE

[Fig. 12]

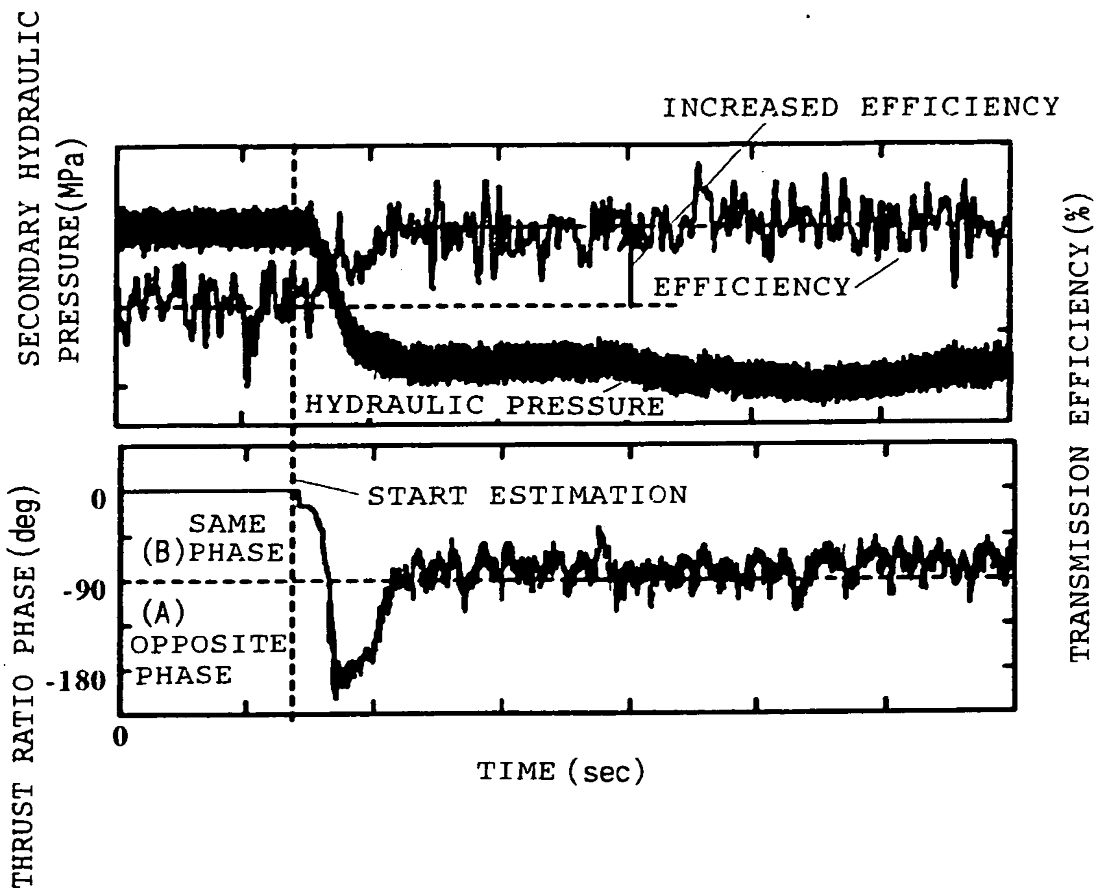

THRUST RATIO
OUTPUT(A)
=SMALL GAIN,
INVERSE PHASE
OUTPUT(B)
=LARGE GAIN,
SAME PHASE
SAFETY VALUE
(SECONDARY HYDRAULIC
PRESSURE)
INPUT(B)
INPUT(A)

[Fig. 13]

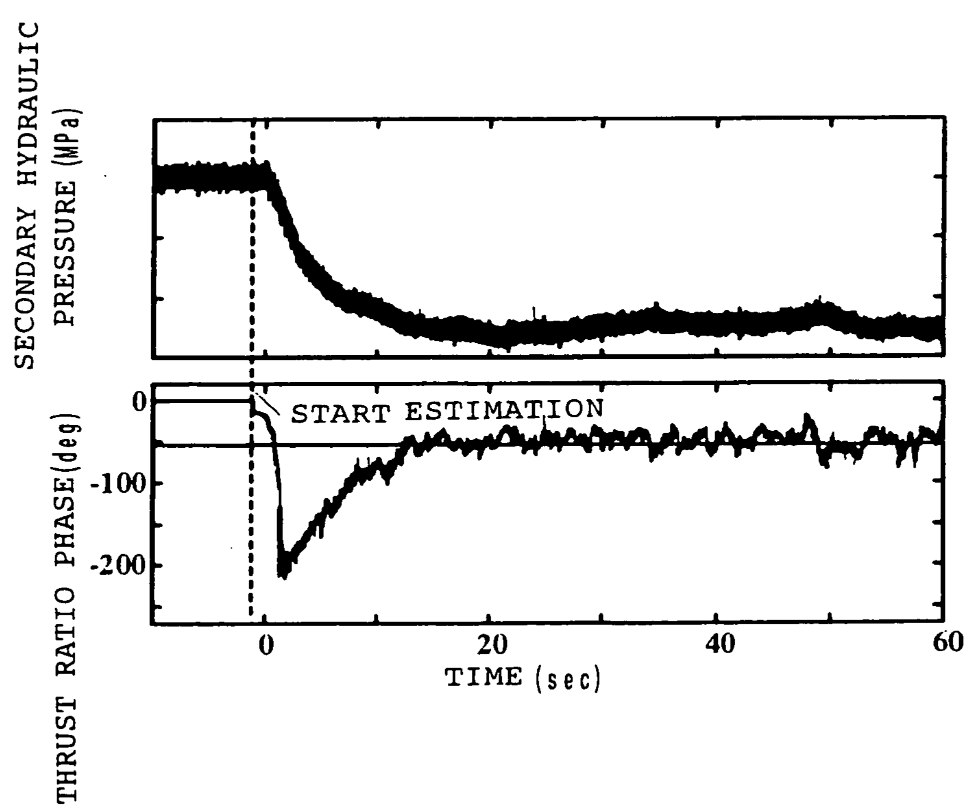


HYDRAULIC PRESSURE-THRUST RATIO TRANSMISSION CHARACTERISTICS

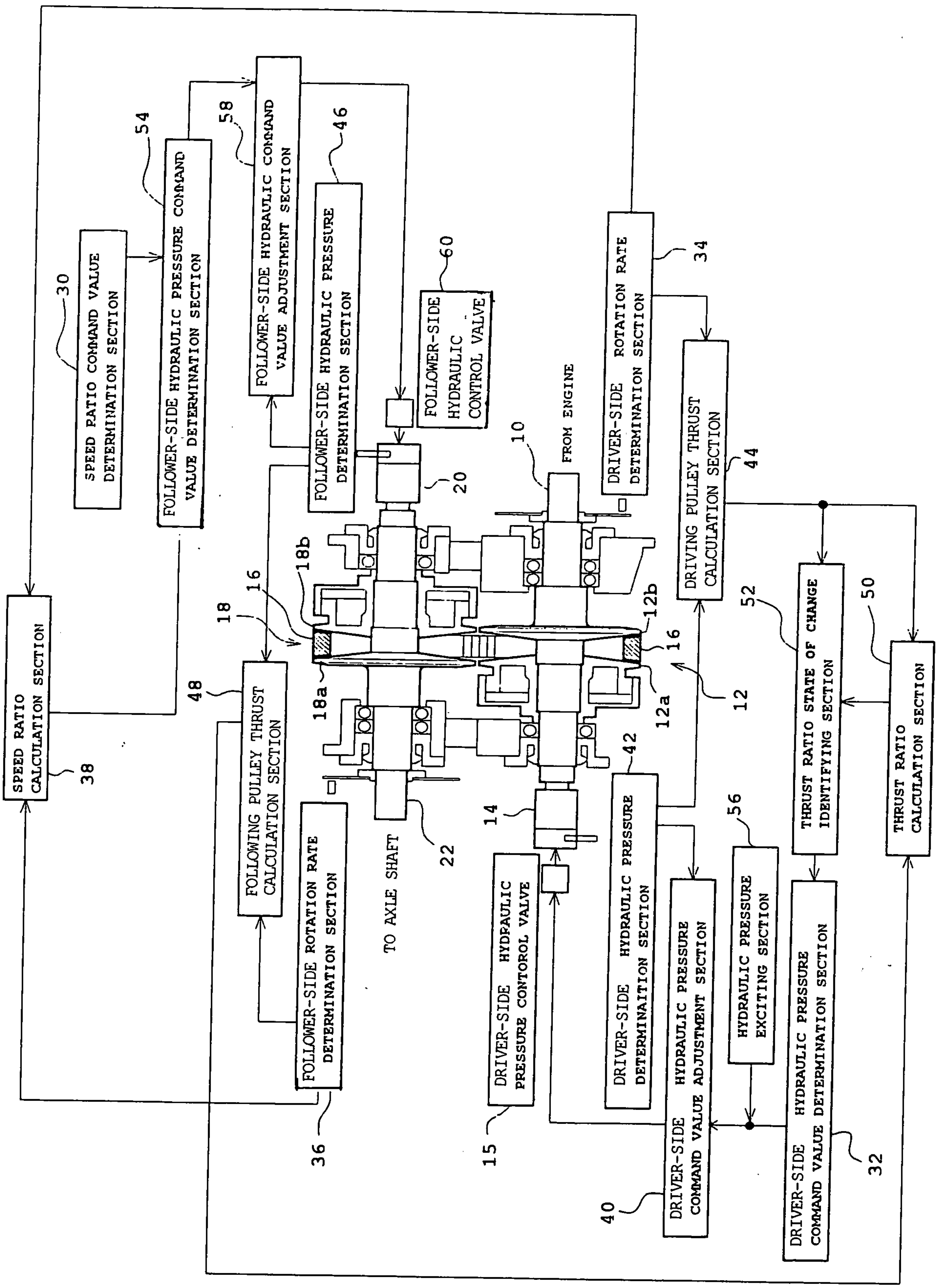
[Fig. 14]


SECONDARY HYDRAULIC PRESSURE(MPa)
INCREASED EFFICIENCY
EFFICIENCY
HYDRAULIC PRESSURE
TRANSMISSION EFFICIENCY(%)
THRUST RATIO PHASE(deg)
0
-90
-180
SAME (B)PHASE
(A) OPPOSITE PHASE
START ESTIMATION
0
TIME (sec)

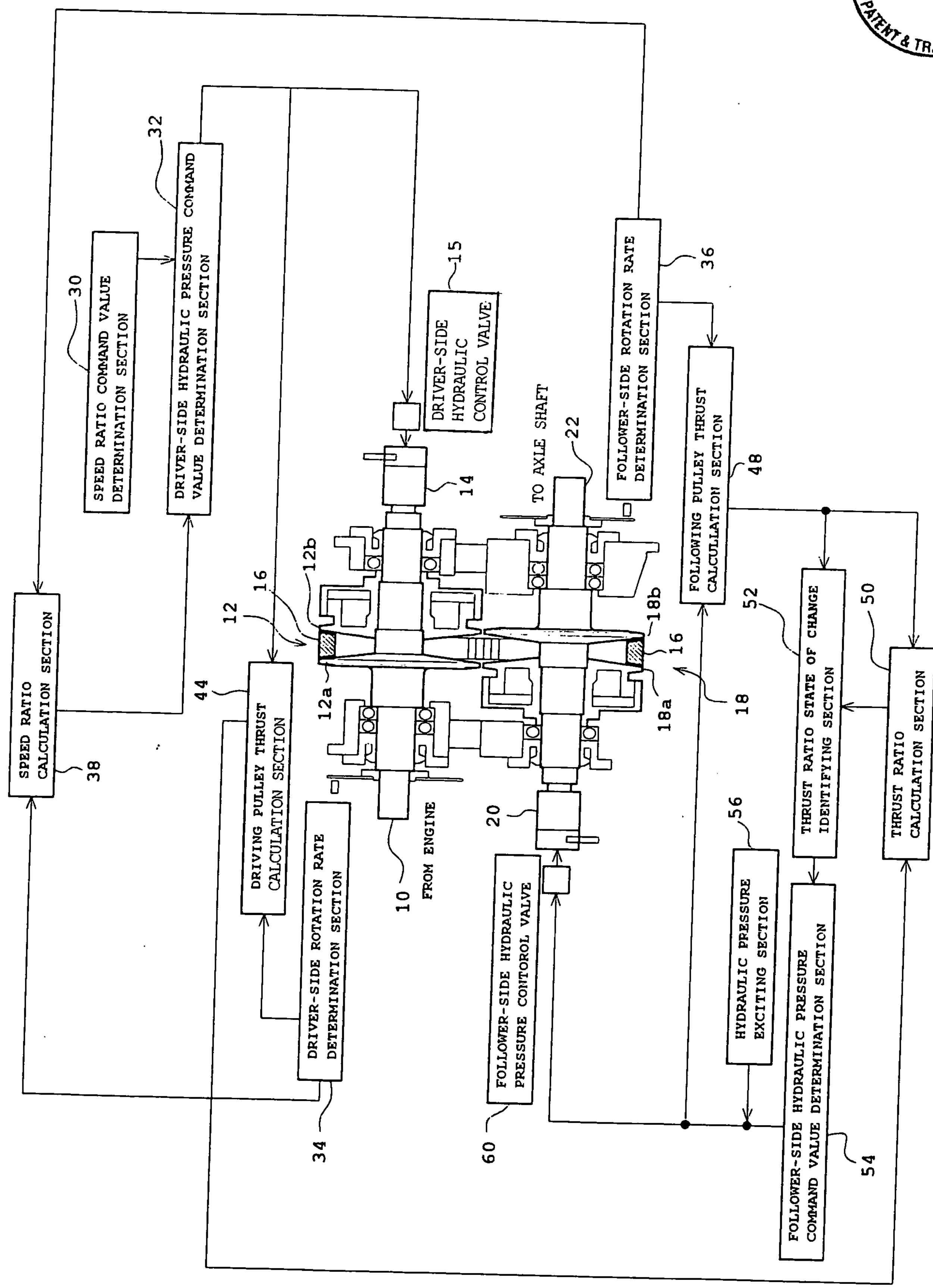
[Fig. 15]


SECONDARY HYDRAULIC PRESSURE (MPa)
THRUST RATIO PHASE(deg)
0
-100
-200
START ESTIMATION
0
20
40
60
TIME (sec)

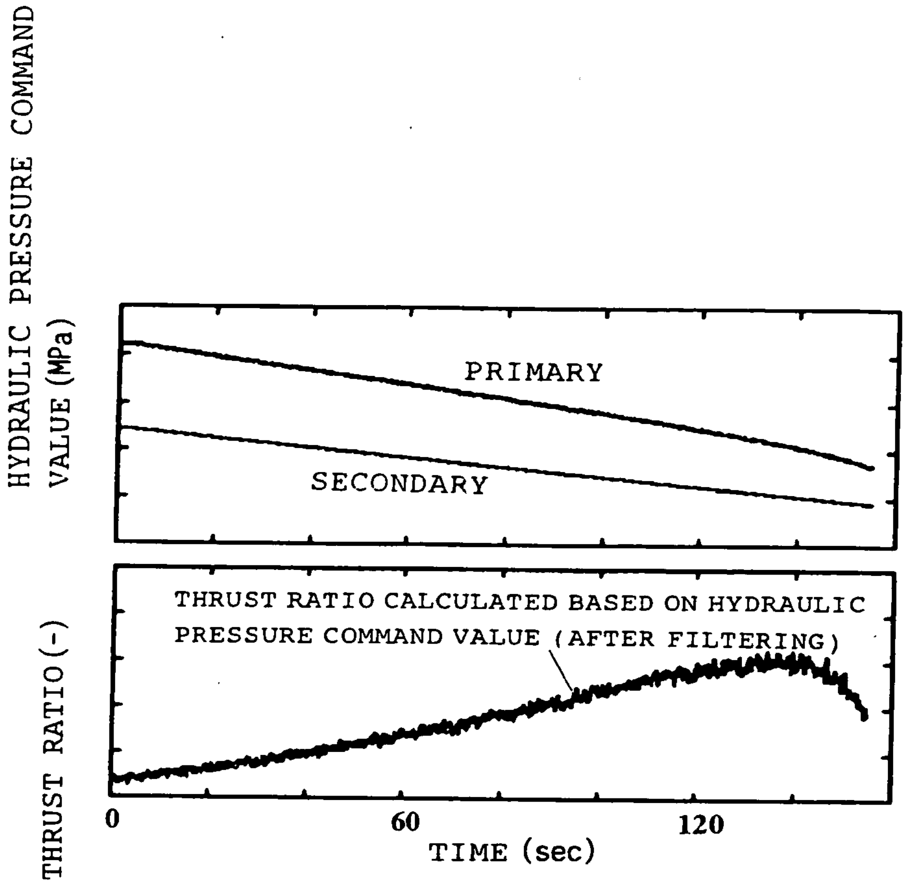
[Fig. 16]


SPEED RATIO COMMAND VALUE DETERMINATION SECTION
30
FOLLOWER-SIDE HYDRAULIC PRESSURE COMMAND VALUE DETERMINATION SECTION
54
FOLLOWER-SIDE HYDRAULIC COMMAND VALUE ADJUSTMENT SECTION
58
FOLLOWER-SIDE HYDRAULIC PRESSURE DETERMINATION SECTION
46
FOLLOWER-SIDE HYDRAULIC CONTROL VALVE
60
FROM ENGINE
10
DRIVER-SIDE ROTATION RATE DETERMINATION SECTION
34
DRIVING PULLEY THRUST CALCULATION SECTION
44
20
18b
16
18
18a
12b
16
12
12a
SPEED RATIO CALCULATION SECTION
38
FOLLOWING PULLEY THRUST CALCULATION SECTION
48
THRUST RATIO STATE OF CHANGE IDENTIFYING SECTION
52
THRUST RATIO CALCULATION SECTION
50
FOLLOWER-SIDE ROTATION RATE DETERMINATION SECTION
36
TO AXLE SHAFT
22
14
DRIVER-SIDE HYDRAULIC PRESSURE CONTOROL VALVE
15
DRIVER-SIDE HYDRAULIC PRESSURE DETERMINAITION SECTION
42
DRIVER-SIDE HYDRAULIC PRESSURE COMMAND VALUE ADJUSTMENT SECTION
40
HYDRAULIC PRESSURE EXCITING SECTION
56
DRIVER-SIDE HYDRAULIC PRESSURE COMMAND VALUE DETERMINATION SECTION
32

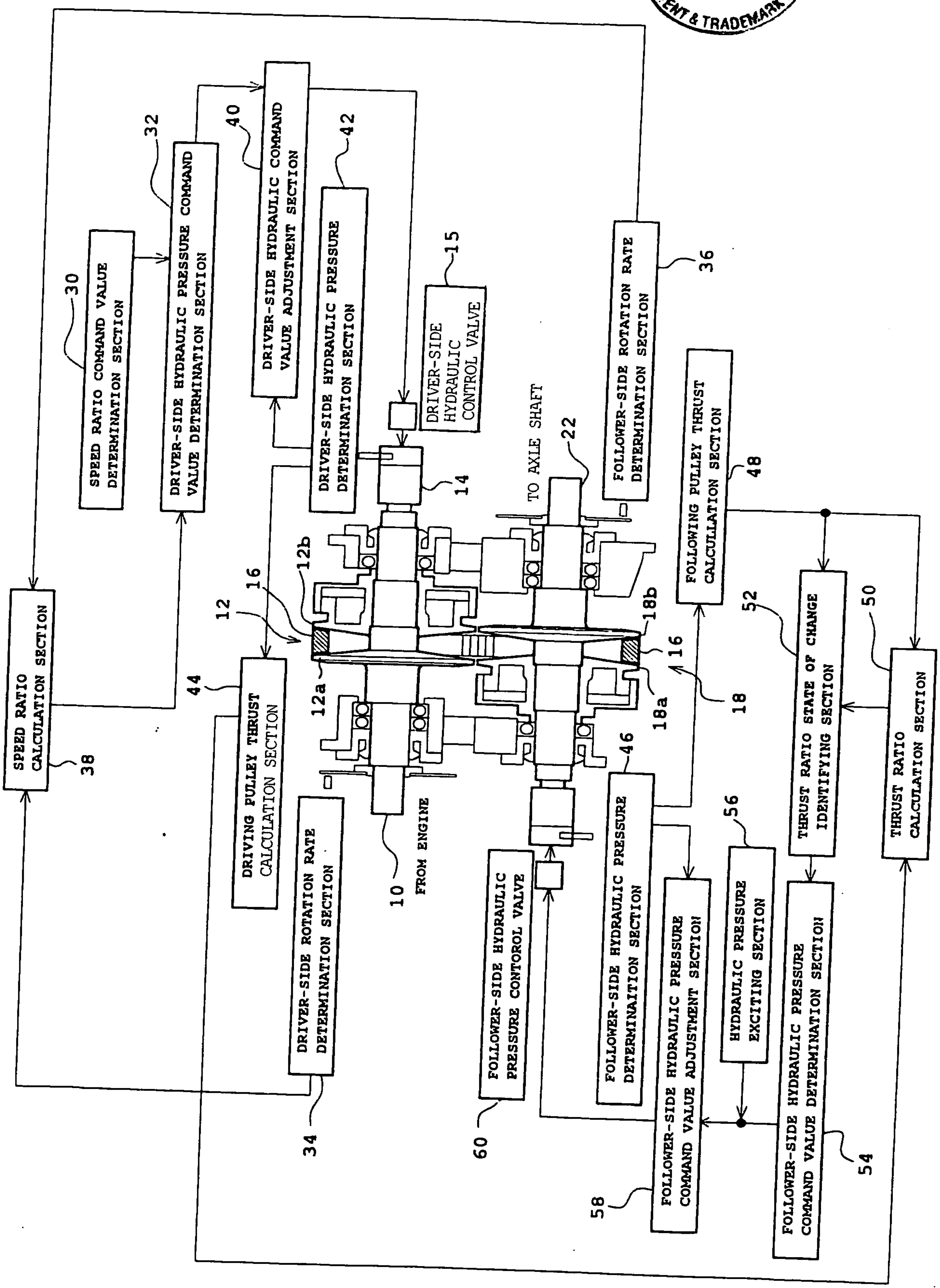
[Fig. 17]


SPEED RATIO COMMAND VALUE DETERMINATION SECTION
30
DRIVER-SIDE HYDRAULIC PRESSURE COMMAND VALUE DETERMINATION SECTION
32
DRIVER-SIDE HYDRAULIC CONTROL VALVE
15
FOLLOWER-SIDE ROTATION RATE DETERMINATION SECTION
36
FOLLOWING PULLEY THRUST CALCULATION SECTION
48
TO AXLE SHAFT
22
14
12b
16
12
12a
18b
16
18a
18
52
THRUST RATIO STATE OF CHANGE IDENTIFYING SECTION
50
THRUST RATIO CALCULATION SECTION
SPEED RATIO CALCULATION SECTION
38
44
DRIVING PULLEY THRUST CALCULATION SECTION
DRIVER-SIDE ROTATION RATE DETERMINATION SECTION
34
10
FROM ENGINE
20
FOLLOWER-SIDE HYDRAULIC PRESSURE CONTOROL VALVE
60
56
HYDRAULIC PRESSURE EXCITING SECTION
FOLLOWER-SIDE HYDRAULIC PRESSURE COMMAND VALUE DETERMINATION SECTION
54

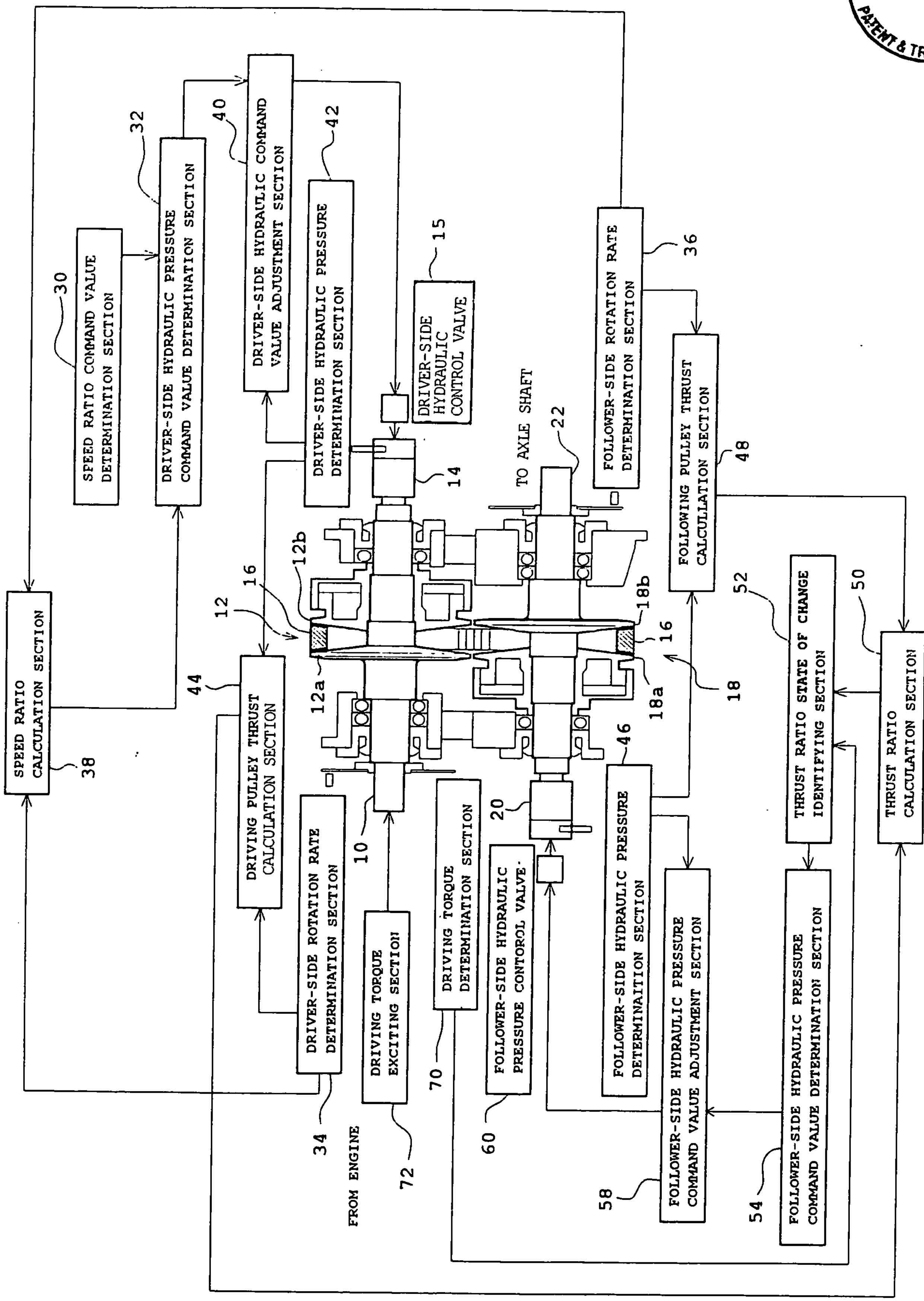
[Fig. 18]


HYDRAULIC PRESSURE COMMAND VALUE (MPa)
PRIMARY
SECONDARY
THRUST RATIO (-)
THRUST RATIO CALCULATED BASED ON HYDRAULIC PRESSURE COMMAND VALUE (AFTER FILTERING)
0
60
120
TIME (sec)

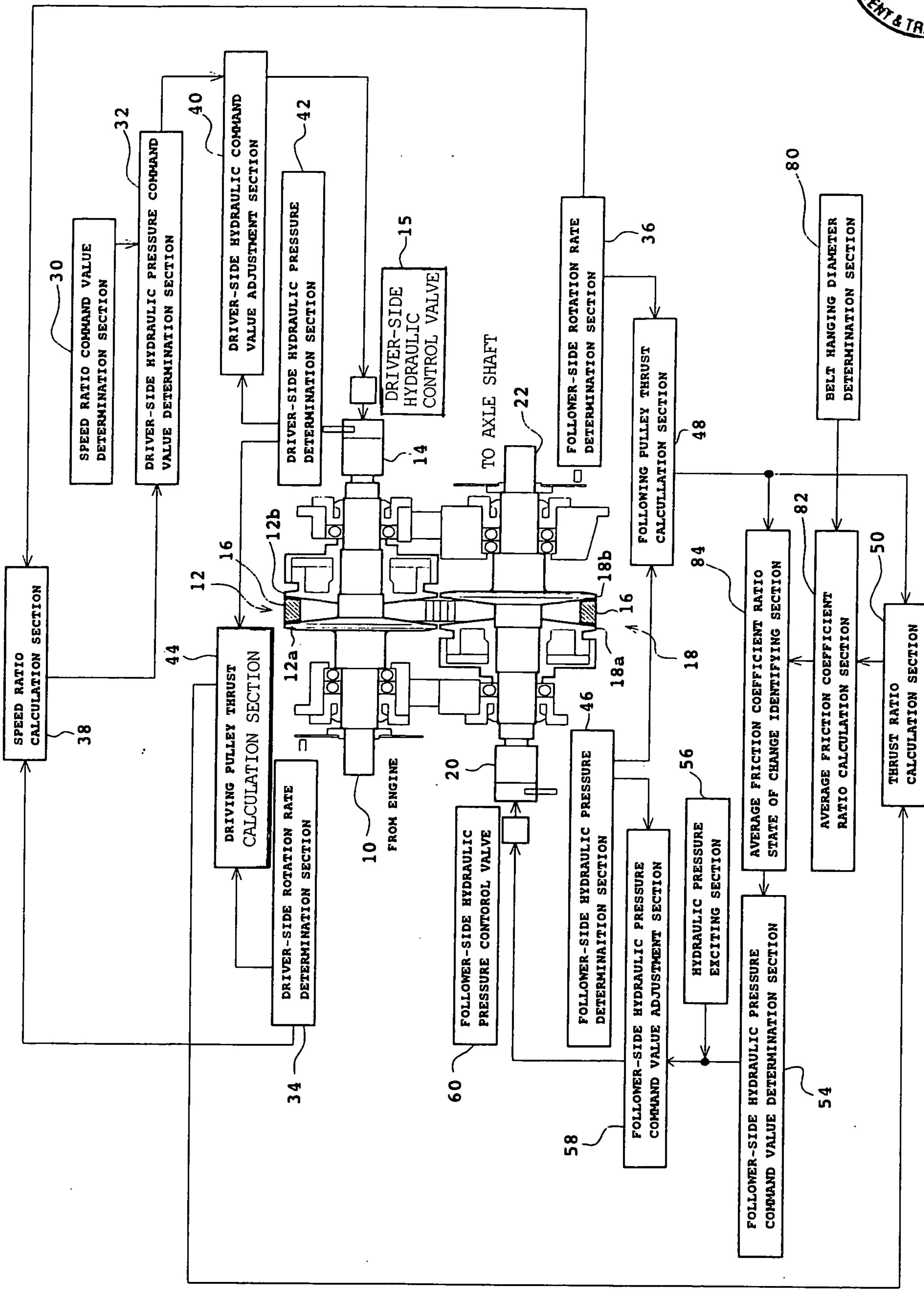
[Fig. 19]


SPEED RATIO COMMAND VALUE DETERMINATION SECTION
30
DRIVER-SIDE HYDRAULIC PRESSURE COMMAND VALUE DETERMINATION SECTION
32
DRIVER-SIDE HYDRAULIC COMMAND VALUE ADJUSTMENT SECTION
40
DRIVER-SIDE HYDRAULIC PRESSURE DETERMINATION SECTION
42
DRIVER-SIDE HYDRAULIC CONTROL VALVE
15
14
TO AXLE SHAFT
22
FOLLOWER-SIDE ROTATION RATE DETERMINATION SECTION
36
FOLLOWING PULLEY THRUST CALCULATION SECTION
48
SPEED RATIO CALCULATION SECTION
38
16
12b
12
12a
18b
16
18a
18
52
THRUST RATIO STATE OF CHANGE IDENTIFYING SECTION
50
THRUST RATIO CALCULATION SECTION
44
DRIVING PULLEY THRUST CALCULATION SECTION
46
FOLLOWER-SIDE HYDRAULIC PRESSURE DETERMINAITION SECTION
56
HYDRAULIC PRESSURE EXCITING SECTION
DRIVER-SIDE ROTATION RATE DETERMINATION SECTION
34
10
FROM ENGINE
FOLLOWER-SIDE HYDRAULIC PRESSURE CONTOROL VALVE
60
FOLLOWER-SIDE HYDRAULIC PRESSURE COMMAND VALUE ADJUSTMENT SECTION
58
FOLLOWER-SIDE HYDRAULIC PRESSURE COMMAND VALUE DETERMINATION SECTION
54

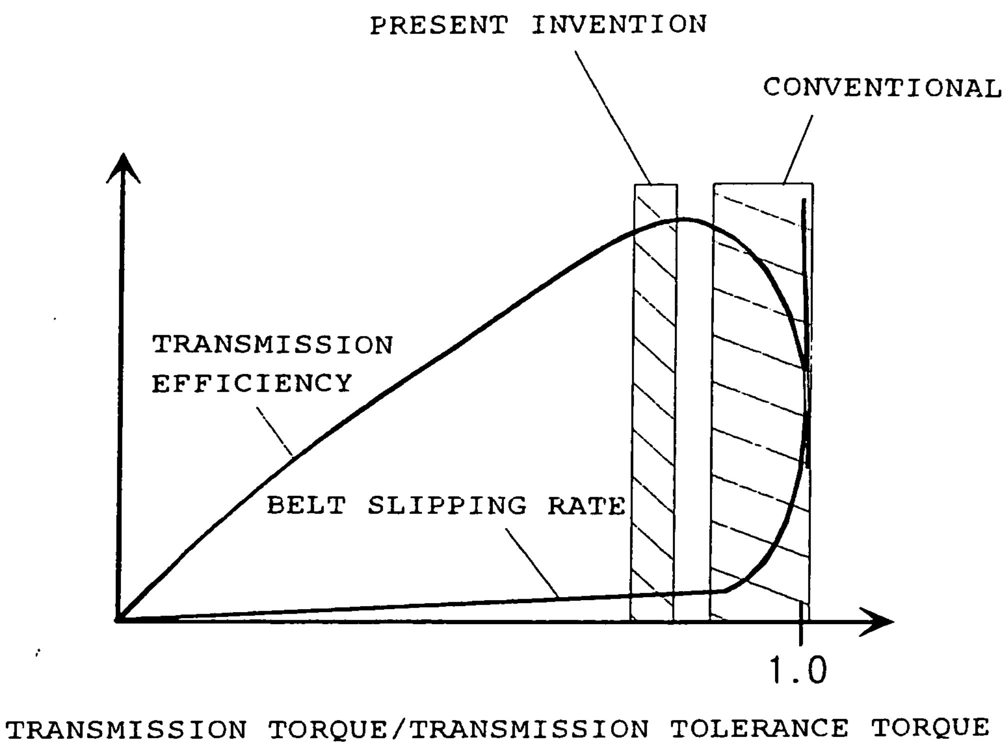
[Fig. 20]


30
SPEED RATIO COMMAND VALUE DETERMINATION SECTION
32
DRIVER-SIDE HYDRAULIC PRESSURE COMMAND VALUE DETERMINATION SECTION
40
DRIVER-SIDE HYDRAULIC COMMAND VALUE ADJUSTMENT SECTION
42
DRIVER-SIDE HYDRAULIC PRESSURE DETERMINATION SECTION
15
DRIVER-SIDE HYDRAULIC CONTROL VALVE
14
TO AXLE SHAFT
22
FOLLOWER-SIDE ROTATION RATE DETERMINATION SECTION
36
FOLLOWING PULLEY THRUST CALCULATION SECTION
48
38
SPEED RATIO CALCULATION SECTION
44
DRIVING PULLEY THRUST CALCULATION SECTION
12
16
12b
12a
18b
16
18a
18
52
THRUST RATIO STATE OF CHANGE IDENTIFYING SECTION
50
THRUST RATIO CALCULATION SECTION
46
20
10
DRIVER-SIDE ROTATION RATE DETERMINATION SECTION
DRIVING TORQUE EXCITING SECTION
DRIVING TORQUE DETERMINATION SECTION
FOLLOWER-SIDE HYDRAULIC PRESSURE CONTOROL VALVE
FOLLOWER-SIDE HYDRAULIC PRESSURE DETERMINAITION SECTION
FOLLOWER-SIDE HYDRAULIC PRESSURE COMMAND VALUE ADJUSTMENT SECTION
FOLLOWER-SIDE HYDRAULIC PRESSURE COMMAND VALUE DETERMINATION SECTION
70
34
FROM ENGINE
72
60
58
54

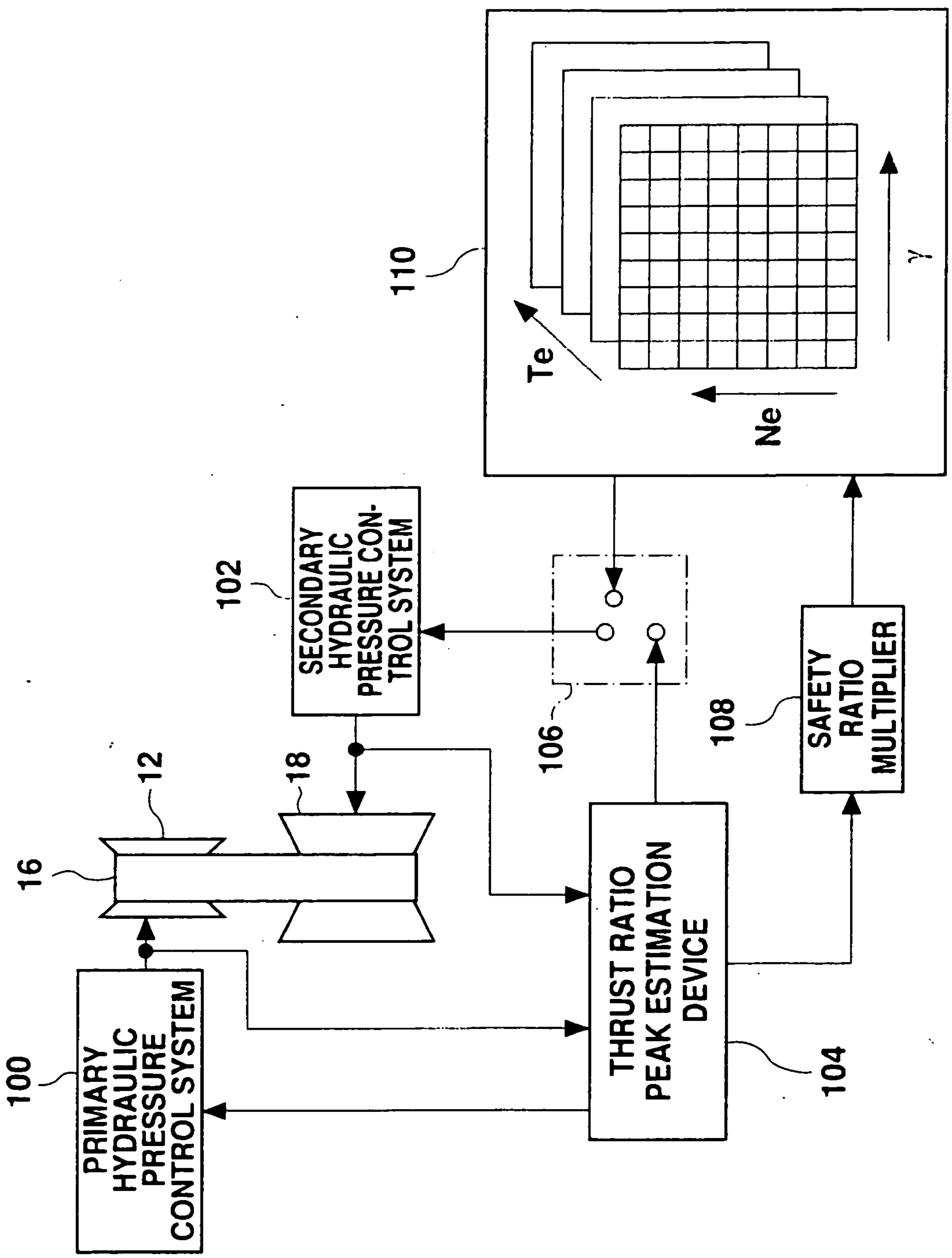
[Fig. 21]


SPEED RATIO COMMAND VALUE DETERMINATION SECTION
30
DRIVER-SIDE HYDRAULIC PRESSURE COMMAND VALUE DETERMINATION SECTION
32
DRIVER-SIDE HYDRAULIC COMMAND VALUE ADJUSTMENT SECTION
40
DRIVER-SIDE HYDRAULIC PRESSURE DETERMINATION SECTION
42
DRIVER-SIDE HYDRAULIC CONTROL VALVE
15
14
TO AXLE SHAFT
22
FOLLOWER-SIDE ROTATION RATE DETERMINATION SECTION
36
FOLLOWING PULLEY THRUST CALCULLATION SECTION
48
BELT HANGING DIAMETER DETERMINATION SECTION
80
AVERAGE FRICTION COEFFICIENT RATIO STATE OF CHANGE IDENTIFYING SECTION
84
AVERAGE FRICTION COEFFICIENT RATIO CALCULATION SECTION
82
THRUST RATIO CALCULATION SECTION
50
SPEED RATIO CALCULATION SECTION
38
12
16
12b
12a
18b
16
18a
18
DRIVING PULLEY THRUST CALCULATION SECTION
44
DRIVER-SIDE ROTATION RATE DETERMINATION SECTION
34
10
FROM ENGINE
20
FOLLOWER-SIDE HYDRAULIC PRESSURE CONTOROL VALVE
60
FOLLOWER-SIDE HYDRAULIC PRESSURE DETERMINAITION SECTION
46
FOLLOWER-SIDE HYDRAULIC PRESSURE COMMAND VALUE ADJUSTMENT SECTION
58
HYDRAULIC PRESSURE EXCITING SECTION
56
FOLLOWER-SIDE HYDRAULIC PRESSURE COMMAND VALUE DETERMINATION SECTION
54

[Fig. 22]


PRESENT INVENTION
CONVENTIONAL
TRANSMISSION
EFFICIENCY
BELT SLIPPING RATE
1.0
TRANSMISSION TORQUE/TRANSMISSION TOLERANCE TORQUE

[Fig. 23]


110
Te
Ne
γ
102
SECONDARY HYDRAULIC PRESSURE CON-TROL SYSTEM
106
108
SAFETY RATIO MULTIPLIER
12
18
16
THRUST RATIO PEAK ESTIMATION DEVICE
104
100
PRIMARY HYDRAULIC PRESSURE CONTROL SYSTEM

[Name of Document] Abstract of the Disclosure

[Summary]

[Problems] To control pulley thrust to an appropriate value.

[Structure] A thrust ratio calculation section 50 calculates a thrust ratio based on driving pulley thrust from a driving pulley thrust calculation section 44 and following pulley thrust from a following pulley thrust calculation section 48. A thrust ratio state of change identifying section 52 detects a peak of change in a thrust ratio in response to changing of the following pulley thrust, based on the thrust ratio and the following pulley thrust. The following pulley thrust is maintained such that the thrust ratio remains at the peak.

[Selected Drawing] Fig. 1